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The Engineering Relations of Shock and Fatigue

The terms "fatigue" and "shock" have been rather loosely used by some engineers. In this article, comprehensive definitions are given and the phenomena described by them are analyzed. There is no need to emphasize its importance.

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IT is a commonplace to say that fatigue and shock constitute two of the most important factors governing the design and performance of the products of the automotive engineer. In any mechanism or structure subjected to rapid movement, the care of the designer and constructor must always be concentrated upon the idea of meeting the demands which the stresses arising directly or indirectly from such movement make upon the various parts exposed to them. The mere dead-weight or steady load rarely throws more than the lightest stresses upon the working parts. The fatigue stresses resulting from rotation or reciprocating motion, and the shocks arising from inequalities of the road or the sudden loading or unloading of the engine, are those which threaten the life of the machine.

This much being well understood, it is perhaps a little surprising to find the terms "fatigue" and "shock" frequently employed in a very loose and vague manner, even by those whose business it should be to understand such matters thoroughly. There is, therefore, some justification for offering a precise exposition of the real meaning of these terms and of the actions to which they refer.

Beginning with the term "fatigue," its real meaning is perhaps best understood from an extreme but very simple example. If a strip of sheet steel be gripped in an ordinary vise, its projecting end can be easily bent over through a right-angle. As a rule it can then be bent back again into the opposite position, through an angle of 180 degrees, and if the steel is very tough and ductile, and the jaws of the vise are rounded to a moderate radius, the strip may be bent backward and forward a number of times. At last, however, it breaks off. This final fracture, after the repetition of a number of similar actions, no one of which—BY ITSELF—would have produced fracture, is a typical example of fatigue. In this somewhat crude example, however, the material has been very severely stressed at each reversal and, consequently, failure has occurred after a very few reversals. If we reduce the severity of the treatment at each operation, the number of reversals required to bring about fracture increases rapidly, mounting up from tens to hundreds and thousands as the angle of bend is diminished. If the process is carried far enough,

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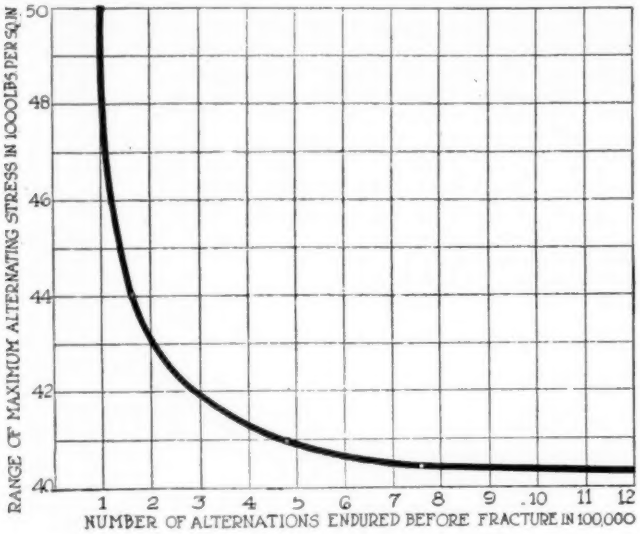


Fig. 1

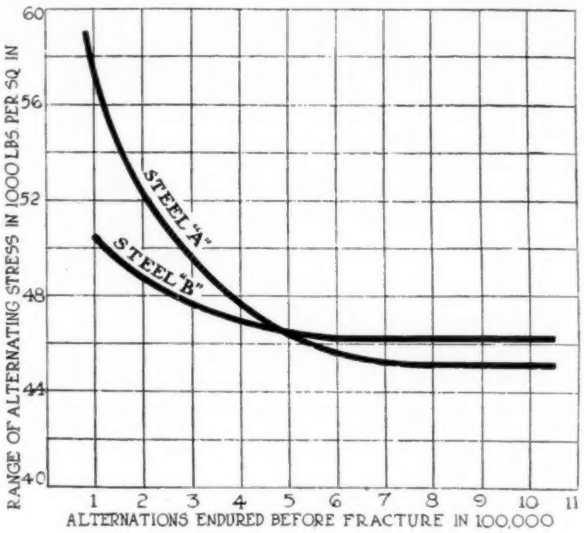


***Fig.* 2**

we reach a stage where the angle of bend is so small that there is no appreciable permanent "set" at each reversal, and yet failure occurs—but only after several hundred thousand reversals. Finally, if the stress applied at each operation is still further reduced, a point is reached where no failure occurs, even after many millions of reversals.

Operating in this manner on a flat strip held in a vise is not, however, a convenient way of making such a test. The most convenient way is to prepare the specimen to be tested in the form of a carefully turned circular rod—preferably hollow or tubular—and to mount this as a cantilever to whose free end a load is attached, generally in the form of a suspended weight. This weight produces a bending stress in the material, which is resolved into a tensile stress in the upper and a compressive stress in the lower fibers of the section of the test-piece. A cantilever thus loaded will support a stress just less than the breaking stress of the material more or less indefinitely. If, however, the test-piece is so arranged that it can be steadily rotated, so that any given fiber of the section comes, when at the bottom, into compression and then—when it arrives at the top, into tension, the conditions of fatigue loading are set up, and the specimen breaks, after a greater or lesser number of rotations, unless the stress is below a certain, well-defined limit.

The Wöhler Test

This test, which is frequently called the "Wöhler" test, must, however, be used with considerable care. In the first place, it is essential to secure the true alignment of the test-piece and of the bearings in which it is rotated, otherwise stresses greater than those calculated may be imposed on the material. Care, however, is also required in the interpretation of the results. One of the most misleading ways of attempting to utilize this test is that of comparing various materials by loading them with the same intensity of stress and then determining the number of reversals which they will endure before fracture occurs. It does not at all follow that simply because, at one particular intensity of stress, material A withstands 300,000 reversals while material B breaks after 200,000, that therefore A has a greater power of resisting fatigue than B.

This fact will be better realized if it is understood that when such a fatigue test is made upon a series of test-pieces of the same material but working in each test with a different intensity of stress, and the number of reversals endured are plotted against intensity of stress, a series of points are obtained, which lie on a parabolic curve which becomes horizontal as the stress applied is diminished. It is really the stress value at which this curve becomes horizontal which measures the value of the material in regard to its power of resisting fatigue. The meaning of the horizontal branch of this curve is, of course, clear: Its height above the line of zero stress determines the highest stress which the material can bear for an indefinite number of reversals without fracture. A typical curve of this kind is shown in Fig. 1.

The Limiting Safe Range

Now it may very well happen—and in practice it does occur—that the graphs obtained with two different materials cross one another, in the manner indicated in Fig. 2. If one were to attempt a comparison between these two materials by means of tests at one intensity of stress alone, the comparison would be misleading if the stress chosen happened to lie to the left of the point where the two curves cross one another. Fortunately, the true parabolic curve can be determined with sufficient accuracy by means of three tests, giving three points on the curve. In this way, the limiting "safe range" of the material may be determined; this is a datum which no serious designer can afford to ignore—nor can it be safely guessed at or determined from the data afforded by tensile tests or "shock" tests.

The failure of metal under repetition of a stress which, when steadily applied would never produce even serious visible deformation, was for a long time a very mysterious phenomenon, and the use of the term "fatigue" suggests that in the early days it was correlated to some unknown property of the metal which became "tired out" by repeated reversals of stress. Fortunately, however, metallographic research has cleared the matter up, and the mechanism of fatigue failure, at all events in its main features, is now well understood.

When a metal undergoes plastic deformation, such as extension during a tensile test, the individual crystals of which the metal is built up each undergo a change of shape also, and they do this not by a process of plastic flow but by a definite process of gliding or slipping, in which the various layers of each of the crystals slide over one another to a minute extent. Thus the crystals accommodate themselves to the new shape required of them, much

as a pack of cards can be distorted by making the cards slide slightly over one another. But this process of internal slip only begins when the crystal undergoes permanent "set"; so long as the deformation remains purely elastic, no "slip" occurs within the crystals. In an ordinary tensile test, signs of slip become apparent as soon as the limit of proportionality has been passed. If, however, very extended search could be made, an isolated crystal here and there would be found to have undergone a slight amount of slip at stresses below the apparent elastic limit.

If, now, the load is a steady one, the occurrence of even a considerable amount of slip leads to no evil consequences; the metal is slightly altered in shape, but remains as strong as ever. For that reason a load, sufficient to bend a test-piece quite appreciably, is yet supported indefinitely. If, however, the load is an alternating one, a different series of phenomena arise. The slips which occurred in one direction on the first application of the load, become reversed with the reversal of the load, and the gliding surfaces within the strained crystals are thus kept in constant motion over one another. There is good reason to think, however, that even the single gliding motion of one slip is sufficient seriously to disturb the arrangements of the atoms of the metal in the adjacent regions of the crystal. At each reversal of the slip this disturbance is increased in severity and depth, until, after a sufficient number of reversals, the material of the crystal on either side of a surface of slip becomes disintegrated.

When this stage has been reached, the crystal in question ceases to exist, so far as any power of bearing a load is concerned and the stress is consequently concentrated upon its neighbors.

Even if these adjacent crystals had hitherto escaped deformation and slip, they can resist no longer, and these crystals in turn undergo disintegration along surfaces of slip. In this way a crack or fissure is formed, and this creeps gradually from one crystal to the next until the entire piece is broken.

A study of the actual microscopic evidence on which this briefly-summarized statement is based, serves to impress one with the vivid reality of the whole process, while it serves to explain many of the peculiar features always seen in "fatigue" fractures. But for our present purpose we need only consider the one main inference: If fatigue failure is to be avoided, the occurrence of slip, even in an isolated crystal, must be avoided. In other words, the stress must be kept low enough not to exceed the limit of elastic recovery even locally within the metal in the case of crystals which happen to be so situated as to yield easily.

The Apparent Elastic Limit

Unfortunately, the determination of the apparent elastic limit, even by means of a very sensitive extensometer, is no true guide in this matter, while the attempt to infer the value of this true elastic limit from the "yield point" as found in an ordinary tensile test is not at all hopeful. The difficulty lies in the fact that, particularly in the case of materials consisting of crystals which vary widely in size and orientation, yielding may occur in a few isolated crystals while the total deformation of the metal still appears to be truly elastic. For the present, therefore, determination of the "safe elastic range" by means of the Wöhler test seems to be the only sound course. If this is followed, it will be found that in some cases—particularly where a material has been subjected to cold work or to certain forms of hardening—the apparent elastic limit in tension may differ very widely from the "true" elastic limit. Such a result can and must be interpreted to mean that the apparent high elastic limit of such material is in this sense fictitious and of no value from the fatigue point of view.

A word may now be devoted to the bearing of these considerations on questions of design and choice of materials. In the first place, if a part exposed to alternating stresses such—for example—as a rotating axle, a crankshaft or a connecting rod—is to be immune from fatigue fracture, it must be so designed that the alternating stresses lie safely below the true elastic range. This can be done in many cases; sometimes, however, the weight entailed would be prohibitive—as in the case of the crankshafts of most airplane engines. In that case a working stress known to exceed the safe range may be adopted, but only if it is clearly understood that the part in question has a limited life and must be renewed after a definite number of reversals. Design on such lines was, no doubt, necessary for war purposes, but is hardly satisfactory for ordinary commercial purposes.

The Choice of Materials

With regard to the choice of materials, it is evident that if it could be regarded from the point of view of fatigue alone, the selection of material would resolve itself into a search for that having the highest safe range or true elastic limit. This, however, would entail the use of very "hard" materials possessing little or no ductility. Here it is important to realize that from the point of view of pure fatigue, ductility is a positive disadvantage, since the slightest sign of deformation is equivalent to the commencement of rupture. In this connection it may be interesting to recall that the Wöhler test has been blamed because it fails to detect the brittleness of such a material as steel high in phosphorus content; in reality one might equally expect a chemical analysis to detect the presence of blow-holes in an ingot.

It is, of course, at once evident that the choice of material cannot be made, even in parts exposed mainly to alternating stresses, on the basis of fatigue resistance alone, and that the power of resisting "shock" must also be considered. This brings us to the consideration of the meaning and nature of "shock."

Here we have to deal with a much simpler matter than in the case of fatigue. For some time the view was strongly held that the application of "shock" brought into play powers and properties of a material differing widely from those exercised in resistance to steady or so-called "static" loads. There is, however, only very little evidence to support such a view. It has recently been shown that in the case of iron and very mild steel there is a definite difference in the manner in which the crystals undergo deformation when strained by a blow or by a steady load, but this difference relates merely to the formation or non-formation of what are known to metallurgists as "twinned" crystals. Actual impact tests made with blows of varying velocity indicate that within very considerable limits the rate at which the load is applied makes very little difference. When speeds like those which occur in guns and projectiles are concerned, this principle no longer applies, but for most types of engineering service one need not think of "shock" as anything more than the rapid application of a stress—and of its effect as determined by the maximum intensity which the stress reaches. The latter point, however, is the vital one, since it is very materially affected by the nature of the material as well as by the shock applied.

When a "blow" is applied to a piece of metal, the intensity reached by the resulting stress is measured by the negative acceleration required to bring the moving object to rest or to reverse its motion. The magnitude of this acceleration, however, depends upon the distance through which the piece of metal is itself displaced. If, therefore, the metal is hard, with an elastic limit close to its ultimate strength, the stress set up by a violent blow may rapidly rise to the elastic limit, exceed it and cause rupture. On the other hand, a tougher material, having a

lower elastic limit, but considerable power of undergoing deformation and taking up work, behaves in a different manner. The stress again mounts up quickly to the elastic limit, but on exceeding this value the energy is rapidly taken up by a slight plastic deformation of the metal: The part is slightly, perhaps imperceptibly, distorted, but no fracture occurs.

It would seem, then, that for parts which are exposed to severe isolated blows a ductile metal not having an excessively high elastic limit is desirable. This is, of course, in direct contradiction to the requirement for resisting fatigue, and the question arises whether the same part must be prepared to deal with both kinds of stress? This is a point which can only be settled by the careful consideration of each part by the designer, and only a few general considerations can be given here.

In the first place, shock itself may be repetitive and produce fatigue failure. That this is actually the case can be clearly shown by making a series of tests in one of the impact machines designed by Dr. Stanton. In this machine a round test-bar having a groove turned in its middle portion is held upon supports and subjected to a blow which applies a bending stress. The machine then turns the bar round through 180 deg. and the blow is then repeated. After a certain number of these alternating blows the test-bar breaks. Now the nature of this test can be very greatly varied by altering the power of the blow and, thereby, the intensity of stress produced. If the blow is a severe one, so that the bar is appreciably bent each time, failure occurs after relatively few blows, and if different materials are compared under such a test they are found to give the same order of performance as under a single-blow notched-bar test of the more usual type. If, on the other hand, tests are made under various light blows, parabolic curves are obtained precisely similar to those obtained from the Wöhler test, thus showing that the mechanism of fatigue under blows producing stresses not much above the elastic limit is very similar to that resulting from steady alternating stresses.

The designer, then, is confronted with a serious dilemma. To obtain resistance to pure fatigue and to the fatigue effects of repeated blows, he must strive to keep the operative stresses well below the "true" elastic limit—and he would therefore select a hard material possessing a large elastic range. But to resist the occasional really severe blow which would produce excessively high stresses in a hard material, he must employ a metal having a large amount of ductility. Fortunately, while it is important that stresses of the "fatigue" nature, which are regularly repeated many hundreds of thousands of times, should be kept within the elastic range, it is not important, or even particularly desirable, to keep the stresses due to exceptionally severe shocks within that range. This consideration enables the careful designer to select a material suitable to his particular purpose, having an elastic range wide enough to allow him to meet the one set of requirements without excessive weight, and yet having sufficient ductility to meet the countervailing demand for shock resistance.

In regard to both classes of stress, however, the greatest difficulty which confronts the designer is probably that of arriving at any correct estimate of what are the real operating stresses in the various parts of his structure or mechanism. Complicated shapes and complex systems of loading combine to make it extremely difficult if not impossible to arrive at such estimates, and in that case the final result must depend upon the judgment of the engineer. But that judgment can only be sound if it is based upon a true appreciation of what are the types of failure and of resistance which may be anticipated from our engineering materials. The present account of some of the fundamental factors governing the behavior of metals under fatigue and shock may perhaps assist in the formation of such a judgment.

The Graphite Industry and Its Uphill Struggle

"STRUGGLING to its feet" sums up the situation of the graphite industry in the United States. The business is in its infancy here and methods are unmatured. There are almost as many processes as there are mines, so that both scientific and business methods require systematizing and refining. Alabama has by far the largest output, but many of the mines are without railway facilities and adjacent roads are poor. The motor truck and road improvement probably would be the solution in that section of the country.

Other sections where graphite is produced include New York, Pennsylvania, Colorado, California, Montana, Texas and Alaska. It does not seem to be a widely known fact, although the fact is substantiated, that every grade of graphite, including the very best, is produced in the United States. The best grades are consumed largely in the manufacture of crucibles. Other industries requiring smaller amounts of graphite of various grades are the stove polish business, foundry facings, lead pencils, paint and lubricants.

Artificial graphite made from coal or other carbonaceous material in an electric furnace is coming to the front and is considered of superior merit, particularly in the manufacture of electrodes. Blast furnace graphite that separates from pig iron when it solidifies offers a possible source of supply in the future, but as yet methods based on this principle are undeveloped, although flakes thus obtained have given good results in lubrication.

During the last two years of the war the amount of graphite imported into the United States was about eight times the domestic production. Ceylon, the principal source of supply in the past, now shows signs of exhaustion, however, and is likely to restrict exports, while Madagascar products to some extent are being substituted. As a stimulus to the trade, no tariff has been imposed on importations into this country since the year 1872.

Although methods of production in Ceylon are crude, this shortcoming is offset by cheap labor and the rich quality of the graphite beds. Hand windlasses are used for hoisting from mines 100 ft. or more in depth. The mineral as it comes to the surface may have as high as 50 per cent impurities after having been sorted below. Hand picking at the surface brings the material to within 5 or 10 per cent of standard quality, and then shipment is made to the coast. There the graphite is screened and hand sorted again. Women chop up the larger lumps and pick out the coarser impurities, after which the pure material is polished with burlap. Grades from several mines are then blended to conform to the requirements of purchasers.

Having a distinct value in the manufacture of lubricants, paints, foundry facings, electrodes, and crucibles, graphite touches the automotive field at many points.

Gas Tightness Tests of Spark Plugs

As a slight escape of hot gases rapidly raises the temperature of the plug, numerous failures are due to this cause. In studying this effect, the Bureau of Standards has tested several hundred plugs. Conclusions are drawn.

DURING the past year the Bureau of Standards has had occasion to test for gas tightness several hundred spark plugs of a wide variety of designs. In a recent report (of which L. B. Loeb, L. G. Sawyer and E. L. Fonseca are the authors) the results of these tests are summarized and conclusions drawn recorded.

The leakage of gas through a spark plug is negligible so far as the loss of pressure in the engine cylinder is concerned, but is of the greatest importance as regards the operation of the plug. A very slight leakage of the hot gases carries heat up into the body of the plug, rapidly raising its temperature, and thus causing one or another of a number of different types of failure.

The ordinary spark plug consists of three distinct parts, viz., the central electrode, the shell or outer electrode, and the insulating material, which is usually porcelain, mica, or glass. There are thus in general two joints which must be made tight, though in some instances manufacturers have constructed both the central electrode and the insulator in two or more pieces.

Various methods of test are now in use for factory inspection and for judging the relative merits of plugs. The method used by the Bureau of Standards for determining gas leakage may be described as follows: The plugs to be tested are screwed into a pressure bomb, which is then filled with compressed air, while submerged in a bath of oil heated to any desired temperature. The leakage of air through the plug is measured by the displacement of oil in an inverted bell jar placed over the plug. Standard conditions for testing the relative merits of different types of plugs are 15 kg. p. sq. cm. (225 lb. p. sq. in.) air pressure and 150 deg. C. (302 deg. Fahr.) temperature.

In the screw bushing method of assembling the insulator has a shoulder one side of which is seated on a shoulder in the shell, while a bushing is screwed down inside the shell on the opposite side. In order to make this joint tight, a gasket, either of brass, copper-asbestos, or some other soft, heat-resisting material, is used. One plug has a copper gasket against the lower surface of the shoulder and a slotted spring-steel washer between the upper surface of the shoulder and the screw bushing. The result of this construction was that the cylinder pressure caused the spring washer to be compressed, moving the insulator upward from the gasket, which was intended to insure the tightness of the plug. Thus the gas escaped readily, causing a very leaky plug. In two exceptional plugs with mica insulation the screw bushing was partially embedded in the mica by great pressure. In these cases no gasket is used between the insulator and the bushing, nor can they be separated without injury to the insulating material.

The crimped shell is formed by forcing the top edge of the shell over a gasket of some soft material, as brass, copper-asbestos, or aluminum alloy, which rests upon the upper side of the shoulder of the insulator. These plugs give fairly good joints; none were absolutely tight, although in many cases the leakage was very small. This method of assembly has the disadvantage that the plug cannot be taken apart for cleaning without destroying it.

The taper fit is used only on plugs of mica or steatite insulation. It failed in all cases but one to give a perfectly tight joint, this being due, undoubtedly, to the flaky property of the mica, which makes it very difficult to produce a smooth and hard surface that will stand the pressure exerted upon it when assembling the plug. One maker has overcome this difficulty by placing a thin steel jacket over the taper, which in this case is approximately 45 deg. This steel jacket is strong enough to withstand the pressure upon it, thus protecting the mica insulation, and at the same time is flexible enough to conform to the surface of the taper within the shell and form a perfectly tight fit.

The molded-in insulators consist of glass which has been forced between the central electrode and the shell while in the molten state. It adheres to both the electrode and the shell, and forms an absolutely tight plug. Attempts have been made to use the molded materials, such as Bakelite and Condensite, in spark plugs, but although these may produce a tight joint when cold, the samples failed to withstand the temperature obtaining in service.

The spun gland is as yet in the experimental stage. It uses a porcelain insulator, the shoulder of the porcelain being surrounded by a spun brass gland which is part of the screw bushing and projects slightly below the shoulder of the plug. The interior of the shell has a taper of 45 deg., which comes in contact with the edge of the gland. When the bushing is screwed down in the first assembly the edge of the gland is crimped about the shoulder of the porcelain, forcing a tight joint. This plug has the advantage of being easily taken apart for cleaning, and can be reassembled to give as tight a joint as at first without the aid of special wrenches. When the plug has once been assembled and the gland formed the insulator and bushing can be separated without totally destroying the gland.

In the majority of cases the central electrode is of uniform diameter throughout and clamped in the insulator. In some instances both the electrode and insulator are threaded and screwed together, and a cement is introduced to insure a tight joint. In still other cases the electrode is cemented to the insulator, there being no other means of fastening these two parts together. Other manufacturers fuse or mold the electrode into the insulator.

It appears from the results obtained that the molded insulator is definitely superior to the other types in the matter of gas tightness. The high rating of the spun gland type is somewhat questionable, since all the plugs tested were constructed specially for experimental purposes. It was shown quite conclusively that leakage depends much more upon the workmanship than upon the design.

Glass plugs gave good results and this was due to their being of molded construction. The difference between porcelain and mica is too slight to be given much weight, but it indicated that the laminated structure of the mica does not seriously decrease its gas-tightness.

Increased Drawbar Pull as Related to Rearing in Tractors

A tractor can "rear" only around its rear axle and not around the center line of ground contact. This is the theory upheld by Mr. Heldt in the following article. It opposes the belief that a tractor can turn over backward with its center line of rear wheel contact as an axis, and that drawbar pull causes the rearing. Blocked driving wheels usually are the cause.

By P. M. Heldt

TRACTOR engineers and students of tractor design for a long time have been divided on the question as to the location of the axis of turning when a tractor rears in service, and also as to the effect of an increase in drawbar pull on the weight distribution between front and rear wheels. The present writer wrote an article on this subject, which appeared in these columns nearly a year ago, and he had hoped that the arguments there set forth would convince every reader that a tractor could "rear" only around its rear axle and not around the center line of ground contact. But from articles recently submitted for publication and from a paper presented before the S. A. E. Minneapolis Section it is seen that there are still those who believe that a tractor can turn over backward with its center line of rear wheel ground contact as an axis and that it is the drawbar pull, or rather the reaction to same, which causes the rearing.

There are two theorems of theoretical mechanics on which those holding the above view base their argument. The first is that when a rigid body is at rest all external forces acting on it are in equilibrium, and the other that the moments of external forces acting on a body at rest vanish, no matter around what axis these moments are taken. One of the outside forces acting on the tractor is the drawbar pull reaction, and if we take a moment of this force around the line of ground contact, we get a reverse torque which is supposed to be the cause of the "rearing."

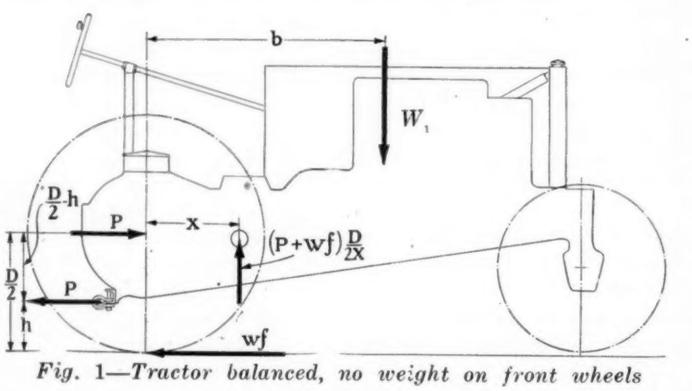


Fig. 1—Tractor balanced, no weight on front wheels

Now, it is absolutely inconceivable that the reaction from an inanimate body like a plow can overturn the tractor. It would be equivalent to raising one's self by his own boot straps. Those who speak of the tractor rearing around the center line of ground contact seem to overlook the fact that this would be reverse rotation of the drivers, and what power should drive the rear wheels in the reverse direction when the motor power tends to cause them to turn in the forward direction? The theorem regarding forces acting on a rigid body should not be applied to the tractor as a whole, because that is certainly not a rigid body. If we hitched a team of horses to the drawbar connection, with the power shut off, and caused them to pull, the tractor would be moved backward but would not turn over.

Usually when a disagreement of this sort occurs it is due to the use of loose language and consequent misunderstanding, one party talking about one thing and the other of something else. In order to avoid this the writer will try to be as explicit as possible.

If we want to apply the law of the equilibrium of external forces acting on a body at rest we should apply it to that portion of the tractor which can be turned around the rear axle or wheel axis; that is, all of the tractor except the rear wheels (if these are free on the axle) or the rear wheels and axle if the tractor is of the live axle type.

First it may be shown that the tractor cannot rear around its wheel-ground contact line. The following argument will, I believe, prove convincing. If the tractor reared on its ground contact line the wheels would be rolling backward. If the center of gravity of the tractor in the normal position is located substantially forward of the rear axle, a considerable amount of energy will have to be stored in raising the forward part of the tractor from the ground. This energy can only come from the engine, and the engine can supply energy only if it turns in its normal direction. But if the engine turns in its normal direction, either the drivers must turn forward or else the frame of the tractor must turn backward relative to the wheels, supposing, of course, a forward set of gears to be in mesh. The wheels cannot possibly turn backward.

In the following let W_1 represent the weight of that portion of the tractor which can swing around the rear wheel axis and b the horizontal distance of the center of gravity of this weight forward of the rear wheel axis. When no power is being applied to the driving wheels this portion of the tractor is held in position by the force of gravity acting vertically downward at its center of gravity, the ground reaction at its front wheels, acting vertically upward, and the bearing reaction of the rear wheels on the axle, vertically upward. We can take moments of these three forces around any axis and they will vanish in each case. For instance, calling the ground

reaction on the front wheels w_f and the wheel base a, then, taking moments around the center line of rear wheel contact—

$$W_1 b - w_f a = 0$$

Now, when power is applied there appear a number of additional forces. There is first the resistance to motion $W f$ (where W is the total weight and f the traction coefficient) acting in a direction opposite to the direction of motion. Of this, however, only a portion, proportional to the actual weight on the front wheels, is impressed upon the frame, and at the moment the front wheels leave the ground in rearing this item obviously becomes zero, and it can therefore be left out of consideration.

Then there is the tooth reaction of the bull gears which has a radial and a tangential component. Let us call the total propulsive effort exerted by the tractor Q, this being made up of the drawbar pull P and the force $W f$ necessary to overcome the traction resistance of the tractor itself. If D is the rear wheel diameter, then the torque reaction or the moment of the tangential component of the tooth pressure around the rear axle is

$$M_t = -\frac{Q D}{2} = -\frac{D}{2}(P + W f)$$

The minus sign signifies that the moment is left hand or counter clockwise. The other moment to be considered is that of the drawbar reaction. If h is the height of the drawbar above ground, then this moment is

$$M_d = \left(\frac{D}{2} - h\right) P = \frac{D P}{2} - P h.$$

The righting moment due to the action of gravity is

$$M_w = W_1 b$$

When the front wheels are just about to leave the ground, these are the only forces acting on the tractor, and the moments must therefore vanish.

$$M_w + M_d + M_t = 0$$

$$W_1 b + \frac{D P}{2} - P h - \frac{D P}{2} - \frac{W f D}{2} = 0$$

$$W_1 b - P h - \frac{W f D}{2} = 0$$

$$W_1 b = W f \frac{D}{2} + P h$$

Therefore, with constant traction resistance encountered by the tractor the rearing moment will increase as the drawbar pull increases and also as the height of the drawbar from the ground increases.

However, the traction resistance is not constant, and there is practically no limit to the value which this factor may assume. We may have, for instance, such a case as that when the drivers are in a ditch and are blocked by a railroad tie or some similar obstruction. The only thing to which there is a limit, and whose limit determines the "rearing" propensities of a tractor, is the rear wheel torque which the engine is capable of developing. The thing that really counts here is not the maximum torque which the engine can produce continuously, but the maximum momentary torque, due perhaps to abstraction of energy from the flywheel. A tractor is naturally most likely to rear when being operated on the lowest gear, and the maximum propelling force which can be obtained under this condition we have called Q. It now remains to see under what conditions the tendency to "rear" is greatest, when all of this propelling force is required to move the tractor itself or when some is applied to the drawbar. In the first case

$$W f = Q,$$

and in the second $\quad P + W f = Q,$

Hence $\quad P = Q - W f.$

In the first case the tractor will rear if

$$\frac{Q D}{2} > W_1 b \text{..................} (1)$$

and in the second, if

$$\frac{W f D}{2} + Q h - W f h > W_1 b \text{.........} (2)$$

Now, the left-hand side of inequality (1) is equal to the left-hand side of inequality (2) if

$$h = \frac{D}{2}$$

That is, if the drawbar is located at axle height the maximum rearing moment is the same whether all propulsive force is needed to overcome the tractor resistance itself of whether some of this propulsive force is applied to the drawbar. If h is less than $D/2$, that is, if the drawbar hitch is below the axle (as is usually the case), then the rearing tendency is greater if all the propulsive force is required to move the tractor than if there is an excess to be passed on to the drawbar, and the difference is the greater the lower the drawbar relative to the axle. On the contrary, if the drawbar is above the axle, then the rearing tendency is greater if some of the propulsive effort is applied to the drawbar. This case needs no further attention because it is practically never met with in practice.

From the above it is plain that in a conventional type of tractor with the drawbar below the axle the tendency to rear is greatest when

(1) The engine delivers its maximum torque;
(2) Power is applied through the lowest forward gear;
(3) All of the torque is required to move the tractor alone or is insufficient to move it.

Hence the above conditions are those which should be considered when investigating the liability of a tractor to turn over backward. If it cannot "rear" under these conditions it cannot do so when pulling a load.

It was above pointed out that it does not matter around what point moments are taken, if the body on which the forces act is at rest the moments vanish. Thus we can take moments around the center line of ground contact (Fig. 1). The righting moment M_w is again the same—

$$M_w = W_1 b$$

The moment of the drawbar pull is

$$M_d = -P h$$

The moment of the torque reaction is

$$M_t = -\frac{D}{2}(P + W f) = -\frac{D P}{2} - \frac{W f D}{2}$$

Then there is a forward thrust P of the wheel on the axle and the moment of this is

$$M_{th} = \frac{P D}{2}$$

Hence at the moment when rearing begins

$$M_w + M_d + M_t + M_{th} = 0$$

$$W_1 b - P h - \frac{D P}{2} - \frac{W f D}{2} + \frac{D P}{2} = 0$$

$$W_1 b = \frac{W f D}{2} + P h$$

which is exactly the same result as reached before. However, since rearing takes place around the rear axle axis, it is a much more natural thing to take moments around that axis.

The Third Brush Regulation in Automobile Generators

The third brush system of regulating generators is commonly used on present cars because of its simplicity and its low initial cost. The following article shows in detail the principles underlying this system of regulation.

By T. R. Knowles

IN order to keep the automobile battery charged there is required a charging source which will give a relatively high current at car speeds of 15 m.p.h. and at the same time will give little, if any, higher currents at speeds four times as great.

With conventional types of generators the output increases with the armature speed. Therefore, when the current must be kept constant over speed ranges of four to one, some special arrangement must be used to regulate the current. The third brush system of regulation is commonly used on present cars, owing to its simplicity and low first cost.

The third brush generator is only a special case of the shunt generator, and one must understand the general principles of the latter before attempting to master the special case.

Fig. 1 is a typical load characteristic of a shunt generator operated at constant speed. It shows that at 1000 r.p.m. the generator in question will deliver 10 amps. at 24 volts and 16 amps. at 20 volts. The curve shows that it is not possible to get 25 amps. at 20 volts, nor is it possible to get 15 amps. at 20 volts. Inspection also shows that the current reaches a maximum at a certain voltage and decreases for lower voltages. The turning back of the curve is not due to armature reaction; the curve would be of the same general shape if there were no other loss than that due to armature resistance.

Curves *A* and *B*, Fig. 2, are the charging characteristics of a three-cell, 150 amp.-hr. battery for different states of charge. Eight volts applied to the terminals of the 1280 s.g. battery will put in 12 amps., while 6.3 volts will send 12 amps. into the 1150 s.g. battery. The points of intersection with the generator curve show that at 1000 r.p.m. the charging rate will be 27 amps. for the full battery and 20 amps. for the empty one. This brings us to the first of the general properties common to all third brush generators: They send current through a fully charged battery at a higher rate than through an empty one. If it were possible to use the upper portion of the generator characteristic (*K*, Figs. 2 and 3) in charging a battery, the more desirable condition of a decreased rate as the battery became charged would be obtained. But it is impossible to use this part of the curve, because, as shown in Fig. 3, the third brush gives regulation only on the lower part of the curve.

Fig. 3 shows an output at 2000 r.p.m., or 13 amps. at 8 volts. Why should the output not be 100 or 200 amps.?

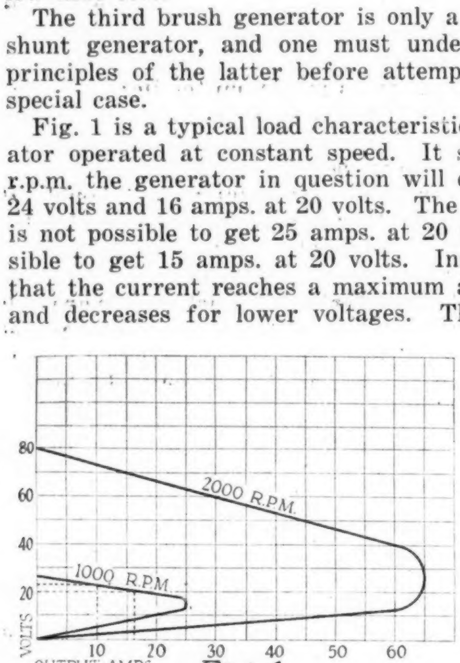


Fig. 1

Characteristic of shunt generator at 1000 r.p.m. and at 2000 r.p.m.

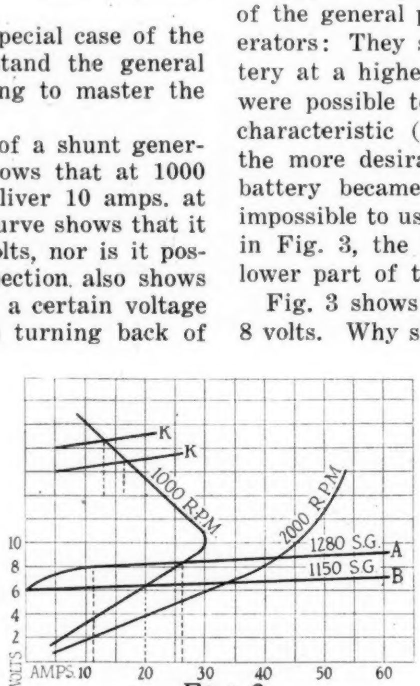


Fig. 2

Points of intersection give the charging rates

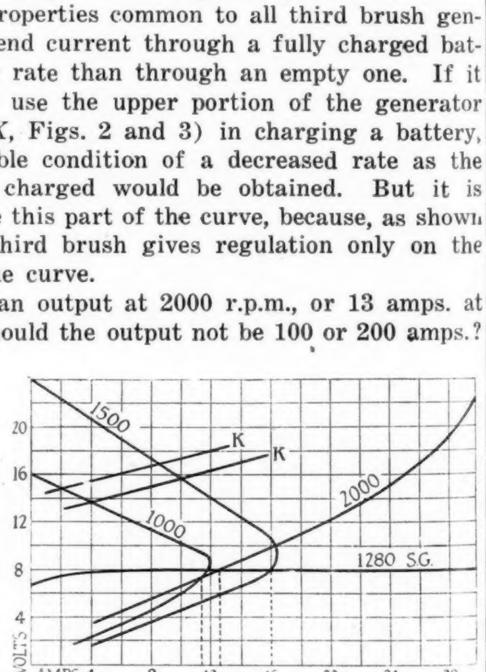


Fig. 3

Characteristics, third brush generator, also battery

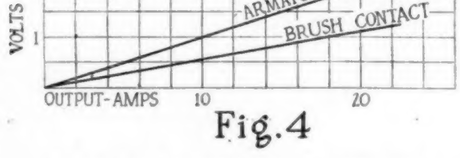


Fig. 4

Volts loss in resistance. Constant temperature assumed

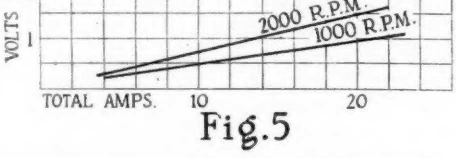


Fig. 5

Reactance volts loss (load current plus short circuit current)

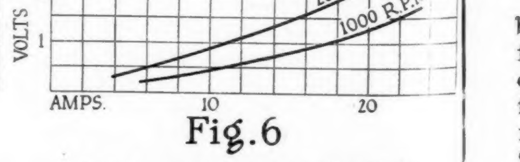


Fig. 6

Total useful flux is reduced by armature ampere-turns

There are five factors causing losses which reduce the output, as follows:

1. Voltage loss due to armature resistance.
2. Voltage loss at brush contact and in brush leads.
3. Voltage loss from reactance.
4. Loss due to field distortion.
5. Loss due to position of field brush (third brush).

The following curve sets show experimental values for each of the above as found in a special four-pole machine:

Voltage loss due to armature resistance (the latter measured by a bridge) is shown in Fig. 4. Curve B gives the brush and brush contact loss, which is equal to the volts across the brushes when forcing current through with all commutator bars soldered together. This was found to be constant for all speeds.

The reactance loss, Fig. 5, is due to a voltage set up in the coils undergoing commutation, due to the reversal of the current in these coils. It is opposed to the useful voltage and is obtained experimentally by taking the apparent resistance at various speeds and loadings with the poles

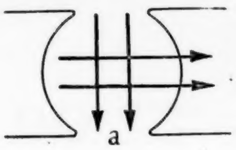


Fig.7

Setting brushes off neutral changes direction of arrows (a)

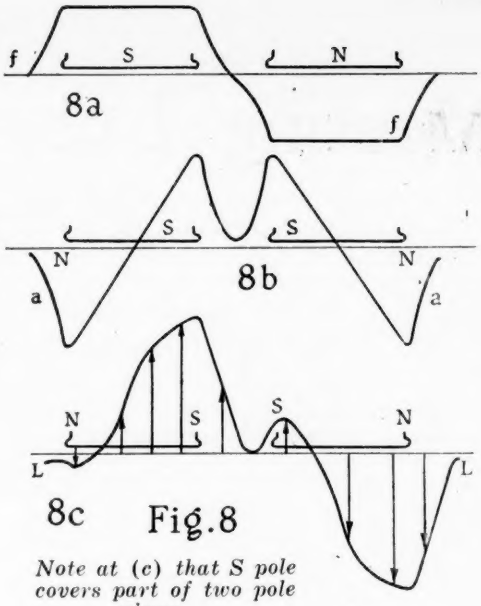


Fig.8

Note at (c) that S pole covers part of two pole pieces

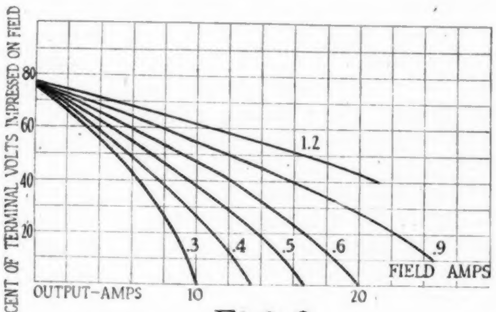


Fig.9

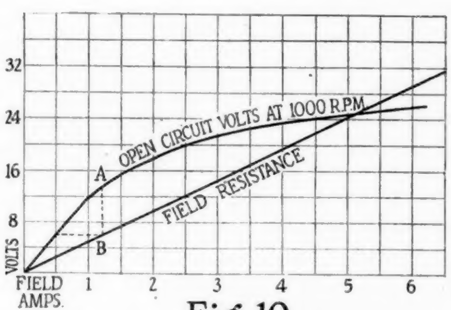


Fig.10

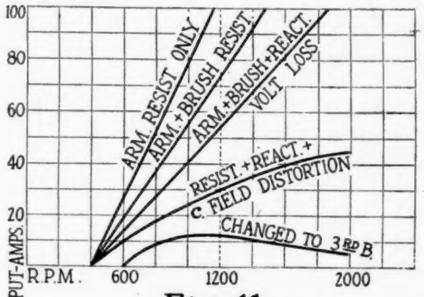


Fig.11

Fig. 9—Curves for one setting of third brush. It is this loss that gives the lower outputs at high armature speeds. Fig. 10—Example: To find output at 6 volts in case only loss is armature resistance. Curve B shows 6 volts will give 1.25 field amperes which in turn will develop 14 volts curve A. Therefore the output will be that current which will cause an internal drop of 14 — 6 volts = 8 volts or 80amperes, Fig. 4. Fig. 11—Output at 8 volts constant temperature assumed. Curve C is output as shunt generator

removed from the frame. Reactance voltage equals the total drop minus the armature and brush contact drop.

With the main brushes set on neutral, the armature ampere-turns cause a flux reduction which gives the voltage loss, Fig. 6. The current in the armature conductors produces a flux, a, Fig. 7, at right angles to the main flux f. Flux a is distributed as shown in Fig. 8b. It adds to the main flux f on one side of the pole, and subtracts from the main flux on the other side. Owing to saturation in the armature iron it subtracts more than it adds and the result is a net loss. This loss is distinct from the loss in field current due to field distortion, and the two must be kept distinct. Fig. 8c shows the resulting field. The line L, usually called the field form, gives a picture of the field strength at all points. The arrows indicate the direction of the flux, and the length of each is a measure of the field strength at that particular point.

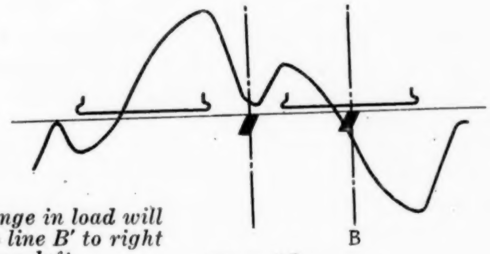
The field form is a function of the field and armature ampere-turns and is independent of the speed. That is, the field shape or field distortion is sensibly the same at all speeds, armature and field current being held constant.

Owing to the position of the third brush, the field does not receive the full terminal voltage. The percentage of the terminal volts impressed on the field, Fig. 9, depends upon the field form. In other words, the data in Fig. 9 may be obtained directly from a set of field form curves such as 8c.

This completes the major losses. There is an additional field set up by the field current flowing in only part of the armature conductors. But its effect is balanced in this machine by the reactance voltage at the third brush.

Properly combining the foregoing with the no-load saturation curve, Fig. 10, Fig. 11 is obtained. From this it may be seen that the generator at 1200 r.p.m. and 8 volts

(*Continued on page 1367*)



Any change in load will shift the line B' to right or left

Fig.12

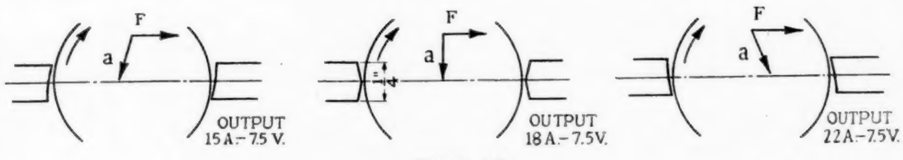


Fig.13

Output changed without changing the third brush. As brushes are given back lead, the armature ampere-turns (a) begin to assist the main field (F)

Meeting Lubrication Requirements in High Speed Engines

In this article, Mr. Brush presents the argument behind his system of high speed engine lubrication. This system has been adopted in whole or part by some of the leading automobile and engine manufacturers. However, discussion of this article is invited, as some of the views expressed do not accord with current practice.

By Alanson P. Brush*

DURING the past decade, the performance characteristics of the automobile engine have been extended until the older lubricating systems do not, and can not be developed to, entirely fulfill modern requirements. A new handling of the lubrication problem is required if the user is to be given the full benefit of this extension of performance.

Eight or ten years ago, in the days of 3-to-1-gear ratios, when a 5-mile stretch of 40-mile-an-hour road was a rarity, when 6 or 7 miles per hour was satisfactory low speed performance on high gear, and when the user was content to shift gears for a ten per cent grade, or even less, the problem of engine lubrication was relatively simple.

With an engine speed range of from 500 to 1500 revolutions per minute, quite ample for normal users' requirements, fairly satisfactory engine lubrication could be obtained in a number of ways. Ordinary splash with pump circulation, automatic circulating splash with a make-up of fresh oil, or ordinary pressure feed with leads to centerline crank shaft bearings and a drilled oil lead from centerline bearings to each crank pin bearing, or various combinations of these systems could be so designed and constructed as to meet quite satisfactorily the needs imposed by these limited engine speeds.

During the past 8 or 10 years, many causes have operated to make correct engine lubrication a much more difficult problem if the user of a motor vehicle is to be given all of the results which he is justly entitled to and rightfully expects.

There are many stretches of road, many miles in length, which, when traffic conditions and road patrols permit, may be safely driven at speeds above 60 miles per hour.

There are many well paved grades, miles in extent, which may be climbed at high speed if the motor will sustain its maximum power output. In coasting these same grades, it should be possible to use the internal friction of the engine to limit the speed of the car instead of wearing out brakes for the same purpose. Congestion of traffic makes high gear speed as low as 2 or 3 miles an hour necessary to the driver's comfort and convenience, and the same condition makes very desirable a high rate of acceleration on top gear at all speeds. Competition has made the demonstration of these various motor car capacities a valuable selling factor.

The up-to-date motor vehicle engine needs a usable speed range of from 100 revolutions per minute or less, to 3000 revolutions per minute or more. It should be able to run indefinitely with little or no load throughout such a speed range and without any perceptible excess of oil in the combustion chambers. If it fails in this requirement, fouled spark plugs, carbonized exhaust valves, objectionable carbon deposits in combustion chambers, excessive consumption of lubricating oil, smoking exhaust outlets and frequent overhaul are among the objectionable consequences to the user.

The modern motor vehicle engine should be equally able to sustain its maximum torque output throughout this wide speed range without the under-lubrication of any part, and if it fails in this respect, undue wear or complete destruction of the working surfaces will result.

Usable engines that approximately meet these requirements are being produced as the volume production of motor vehicles attests, but that these requirements are not satisfactorily or fully met is equally evident.

If this statement seems questionable, examine the records as to spark plug replacement sales, spark plug cleaning, engine overhaul for carbon deposit cleaning and valve regrinding, consumption of lubricating oil, advertisements of special piston rings, piston constructions and other devices to prevent over-lubrication, and, last but not least, consider the recently established practice of automobile manufacturers of cautioning users not to attempt to secure even approximately maximum engine performance until the tightly fitted pistons and cylinders have worn each other enough to produce proper working clearances.

As a matter of fact, and as this article proposes to demonstrate, the problem of lubrication has been so solved that a new engine may be constructed with proper working clearances for sustained power output throughout the widest possible speed range and yet never show perceptible evidence of over-lubrication under any condition.

Every engineer knows that crank pin and crank shaft bearing loads which are due to the motion of the moving parts increase as the square of the speed, and that in modern high speed engines these loads greatly exceed the bearing loads imposed by the explosion pressures in the combustion chambers.

This means that the duty imposed upon the lubricant in crank pin and crank shaft bearings is also increased in proportion to the square of the engine speed, and this duty is actually at its maximum at top engine speed with a zero or negative power output from the engine, that is: The maximum lubrication duty is imposed upon crank shaft and crank pin bearings at a time when over-lubrication of pistons and cylinders is most liable to occur.

The new solution of this problem of lubrication meets this condition fully by providing means for adequately lubricating and cooling crank shaft and crank pin bearings

*Brush Engineering Association, Detroit, Mich.

when under maximum load without throwing off enough oil from the crank pin bearings to over-lubricate pistons and cylinders when their lubrication requirements are at the minimum. By making the crank shaft itself an oil line, and circulating through it a large excess of oil over the amount required for lubricating the crank shaft and crank pin bearings, the crank shaft temperature is kept down to substantially the temperature of the oil in the reservoir. When this is done, adequate lubrication of the crank shaft and crank pin bearings for their maximum load and speed condition is fully accomplished so long as oil is maintained between the bearing surfaces.

With an oil cooled crank shaft there is no necessity of forcing the oil into the bearing with sufficient pressure to cause any considerable throw-off from the bearing. It has been found that a rigidly round bearing without oil grooves or clearances at the part line can be kept full of oil with very little throw-off of excess oil.

It also has been found that with the maintenance of a low temperature of the crankshaft, and consequently of the oil within the bearing, a bearing which is simply a rigidly round hole can be run much looser than an oil grooved, or, for that matter, any type of bearing where the crankshaft temperature is permitted to rise above that of the oil in the reservoir. This accomplishes the fundamental requirement of lubricating the crankshaft and crank pin bearing for maximum load, without over-oiling pistons and cylinders when their lubrication requirements are minimum.

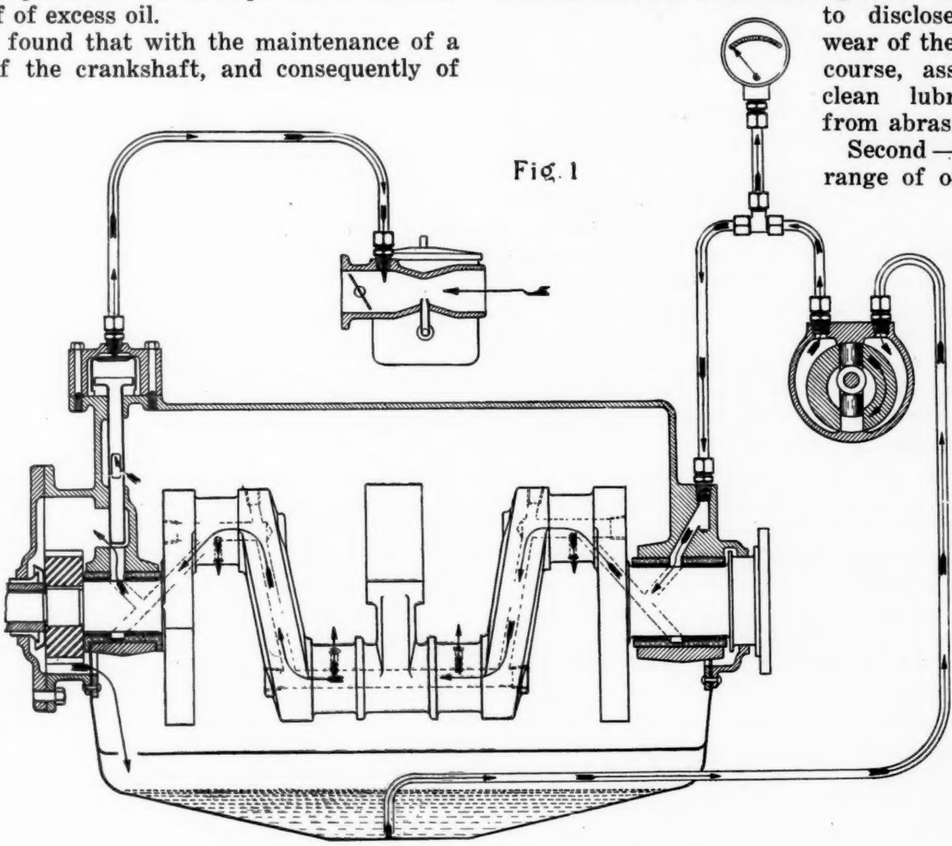


Diagram of complete lubrication system

To meet all conditions of piston and cylinder lubrication correctly, it is only necessary to obstruct the outlet of the oil stream from the crankshaft so as to cause pressure within the crankshaft oil line in proportion to the power output of the engine, that is, to vary the pressure within the crankshaft oil line approximately in proportion to the lubrication needs of the pistons and cylinders.

This is only another way of saying that crankshaft and crank pin bearings cannot be over-lubricated, and that with this system of lubrication, the throw-off from the crank pin bearings may, by varying the pressure of the oil line, be varied in proportion to the piston and cylinder requirements without, in any way, affecting the lubrication of these crankshaft and crank pin bearings.

With this system of lubrication, it is obvious that the internal friction of the engine may be used as a brake for coasting long grades with safety to the crankshaft and crank pin bearings—since their lubrication is always adequate for maximum load—without over-lubricating the pistons and cylinders at this zero or negative power output, and that conversely, the motor may be run at sustained power output without danger to piston and cylinder lubrication, if pressure in the oil line is raised sufficiently to force an excess of oil through the crank pin bearings adequate for maximum piston and cylinder requirements.

With this system of lubrication, new engines may be assembled with adequate working clearances throughout so that they are ready for maximum duty when they leave the assembly stands, eliminating all need of "petting" in the hands of testers or users of new cars.

A number of additional and somewhat unexpected advantages have been found to accrue to the user from this system of lubrication.

First—Crankshaft and crank pin bearings, if properly proportioned and of sufficient rigidity in their roundness, actually seem to ride on an unbroken oil film with the result that tests at full load covering over a thousand hours fail to disclose any measurable wear of these parts. This, of course, assumes the use of clean lubricating oil free from abrasives in suspension.

Second — A much wider range of oil viscosity is permissible; contamination of the lubricating oil as a result of incomplete combustion or faulty intake manifold distribution becomes much less serious, and the grade of lubricating oil which may be used is more elastic.

The development and study of this lubrication system discloses the fact that conventional connecting rod lower end designs are not sufficiently rigid and that to secure the almost, if not absolute, freedom from crank pin bearing wear, which is both possible and desirable, it is necessary to greatly increase the rigidity of this part of the connecting rod.

Crankshaft centerline bearings were also found to be somewhat faulty in this same respect, but most of all was it found that conventional crank shaft design, itself, lacked sufficient rigidity to keep the bearing portions of the shaft in alignment with the intended axis of rotation.

This general fault in crankshaft design has been in the past at least partly justifiable because the old type of closely fitted bearings with their high temperature lubricant showed a rapid rise in friction loss as their tangential speed was increased.

With this new type of lubrication no measurable effect is perceptible in friction loss from an increase in bearing velocity. Correct lubrication has literally taken the lid off as far as crankshaft design is concerned, and no excuse remains for flexible shafts with their attendant noisy operation, bell-mounting of bearings, etc.

A somewhat humorous experience has been encountered in adapting this system of lubrication to automotive engines. The engine assembler accustomed to a constant effort to determine just how tight a bearing can be fitted without burning out, is loath to believe that proper oil distribution can be secured without oil grooves, or that an engine will run smoothly and quietly with from two to five thousandths of an inch actual looseness in crankshaft and crank pin bearings.

The conscientious workman will frequently literally beg for permission to make just a little oil groove, or scrape just a little clearance at the part line, or fit the bearing just a little under the specified minimum of two thousandths clearance; or again he will flatly declare that the Engineering Department is insane and that he will be no party to a practice that is foredoomed to failure. This prejudice on the part of the old-line mechanic is not entirely humorous if he can get a chance at over-hauling one of these modern, free-fitted engines without restraining supervision. What he will invariably do to it is more than plenty.

Scarcely less unhappy is the result when an engineer makes a premature assumption that he has fully grasped the idea and is competent to incorporate this system in an engine design.

The accompanying cuts, it is hoped, will clearly illustrate some of the most important essentials, but this article is in no wise intended to be an exhaustive treatise covering all of the details which must be understood if the user is to be given the full benefit of correct engine lubrication throughout.

For example, many engineers labor under the hallucination that an oil pump to be reliable must be immersed in the oil in the reservoir when, as a matter of fact, even a gear type of oil pump can be made absolutely reliable irrespective of its location, although it must be admitted that this requires constant vigilance in the construction and assembly of the gear type of oil pump.

Again, it is quite generally assumed that any construction of oil pump is all right if it is immersed in the reservoir oil when, as a matter of fact, many oil pumps, even when so placed, will not provide sufficient oil delivery for cooling purposes against the pressures necessary for adequate cylinder lubrication.

Or again, many engineers will, through an exercise of pernicious originality devise elaborate methods of oiling the auxiliary parts of the engine frequently involving the use of minute and accurately calibrated delivery holes which are not only liable to stoppage, but entirely unnecessary.

Another common error in design is to needlessly over-lubricate some auxiliary bearing such as a center camshaft being with consequent over-lubrication of the adjacent pistons and cylinders at lower power outputs.

The writer's purpose in this article will have been served if he has made two things apparent.

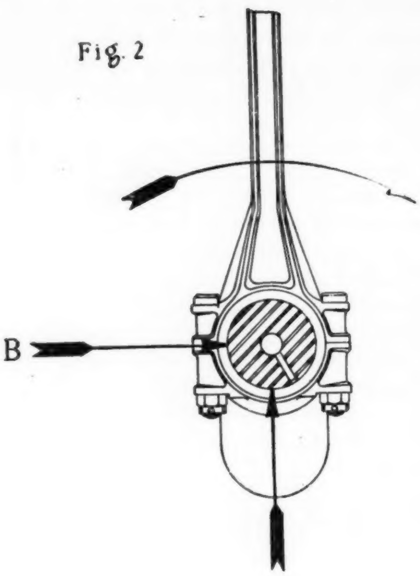
First—That a method of lubricating automotive engines has been devised which will increase the durability and lessen the service needs of modern wide range engines.

Second—That the adaptation of this or any other system of lubrication to such an engine is a highly specialized problem which too often is not given the consideration needed.

Fig. 1 is a diagrammatical illustration of the "Brush" method of oil cooling the crankshaft and regulating oil pressure entirely in proportion to piston and cylinder requirements. A type of oil regulator is shown which is recommended for use with plain tube carbureters. A type of oil pump is shown which is recommended as being more reliable than the ordinary gear pump. Any suitable oil pump in any desired location may be used, also there are a number of other methods of regulating oil pressure in proportion to cylinder and piston needs which are entirely satisfactory. As indicated in the cut, the oil is drawn from the oil reservoir through the pump suction; is forced into the hollow crankshaft at one of the end bearings and allowed to escape from the crankshaft through the opposite end bearing.

With the regulator shown—when the throttle is nearly closed, the oil escapes freely by merely lifting the light piston of the regulator. When the throttle is open the flow of air through the carbureter causes a depression within the regulator chamber and the atmospheric pressure within the regulator cylinder forces the regulator piston down against the escaping oil, thus inducing an oil pressure for piston and cylinder lubrication substantially in proportion to the air velocity through the carbureter which is a sufficiently accurate approximation of the power output of the engine to give correct cylinder and piston lubrication under all operating conditions.

Fig. 2



Crankpin and connecting-rod big end

Figure 2 illustrates the method of oiling each individual crank pin bearing and shows the more rigid type of connecting rod lower end which is necessary if sufficient rigidity is to be secured to eliminate bearing wear.

The oil delivery hole to the bearing should always be located a few degrees ahead of a radical line as shown. This angle should never be more than 45 deg. When an engine is operating at practically zero power output, the load on the connecting rod bearing is substantially radially outward from the center of rotation, and with the location of the oil hole indicated, there is always room for escape of oil from the oil delivery hole into the bearing, if the bearing is rigidly round.

With the oil delivery hole correctly located, when an engine is operating at or near zero power output, the escape of the oil into the bearing will be substantially constant, irrespective of permissible bearing clearance variations. Properly proportioned bearings of this type will operate without knocking with clearances as high as six one-thousandths of an inch, and will oil perfectly with clearances as low as two one-thousandths of an inch.

Since with any increase of power output the mean pressure point on the crank pin moves around *away* from the oil delivery hole, the oil delivery clearance actually increases with the power output of the engine.

S. I. FEKETE and Stuart G. Baits of the Essex Motor Car Co. have been granted a patent on a streamline body in which the nose of the body is closed and conforms to streamline shape in a horizontal plane. The upper and lower portions of the front project ahead and carry the radiator between them. Hence, the radiator is supported ahead of the nose of the body and the air passing through it does not pass over the engine.

Gear Teeth Sizes from the Standpoint of Durability

Part I

All formulæ in current use for calculating gear teeth are based on the breaking strength of the teeth. The author of this article makes the point that in the case of gears for trucks, and especially for farm tractors, the calculations should be based on the resistance to wear of the teeth.

By Joseph Jandasek, M. E., E. E.*

IN designing and proportioning parts of machines which are subject to tension, compression, bending or torsion, the designing engineer uses well-known rules and formulæ in order to determine whether the maximum unit stresses are within certain limits which give a fair margin of safety. To find the necessary dimensions when designing plain cylindrical bearings, we base our calculations on the maximum allowable pressure per sq. in. for each particular case. When this pressure is within the limits found by experience we can safely expect that the bearings will be durable, provided, of course, the material, lubrication, etc., are as specified. However, when this critical pressure is exceeded, the bearing will wear out rapidly. The amount of wear does not increase in the same proportion as the specific pressure, being very small and constant up to a certain value of the pressure; but when the pressure is further increased the wear starts to increase rapidly, until finally a unit pressure is reached with which the bearing has practically no life at all.

Antifriction bearings are designed in a similar way, and it is again the surface pressure between the balls or rollers and the races that determines the capacity of bearings of a given quality of material, workmanship, etc. Before this fact became known the design of ball and roller bearings was entirely by cut-and-try methods; hence the innumerable failures in the early days of antifriction bearings.

The load which breaks a ball or roller is no measure of its actual capacity in a bearing. The crushing load for a hardened steel ball is about one hundred times greater than the working load, the latter depending solely on the surface compression. The well-known formula for the working load of a ball is:

$$P = k\, d^2 \qquad (1)$$

For hard steel balls on flat, conical or cylindrical surfaces and continuous running we have,

$$P = 700\, d^2 \ (P \text{ in lb., } d \text{ in in.}) \qquad (2)$$

which formula is derived from the Hertz equation for pressure between a sphere and a flat plate,

$$C^2 = \frac{0.059\, P\, E^2}{r^2} \quad \text{(Hertz)} \qquad (3)$$

where C = greatest compressive stress in lb. per sq. in.

E = modulus of elasticity; for steel $E = 3 \times 10^7$ lb. per sq. in.

P = compressive force on ball in lb.

r = radius of ball in in.

From the above we readily obtain

$$P = \frac{C^2}{0.059\, E^2}\, r^2 \text{ or}$$

$$P = 4.24\, \frac{C^2}{E^2}\, d^2 \qquad (4)$$

Consequently, the constant k in equation (1) depends only on the quality of material—

$$k = 4.24\, \frac{C^2}{E^2}$$

and is directly proportional to the greatest allowable compressive stress C, the modulus of elasticity E being constant for each particular kind of material.

For roller bearings we have, similarly,

$$P = k_1\, f\, d \text{ lb.} \qquad (1a)$$

where, according to Hertz,

$$k_1 = 2.86\, \frac{C^2}{E} \qquad (4a)$$

The constant k_1 determining the capacity of roller bearings depends only upon the quality of the material (maximum allowable compression C) and upon the modulus of elasticity E (more flexible rollers generally allow of greater loads).

From the above we recognize that in the case of mere sliding action (plain bearings) as well as mere rolling action (antifriction bearings) there exists a certain critical unit pressure beyond which it is not advisable to go if we want the working surface of our bearings to be durable.

Although this theorem that the surface compression determines the capacity of antifriction bearings was proved by numerous tests, it did not gain recognition in everyday engineering practice. The problem of calculating pressures between bodies with curved surfaces and using the greatest compressive stresses as a guide in the determination of the dimensions of those particular machine parts seems to be little understood.

In practice we most frequently meet with the following forms of contact between bodies with curved surfaces:

1. One cylinder rolling on another cylinder (Fig. 1) or on a plane (as for instance camshaft and roller cam follower).

2. One cylinder sliding on a plane (Fig. 2) as, for instance, camshaft and mushroom cam follower).

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*Engineer of the Paige-Detroit Motor Car Co.

All illustrations for this article were prepared by N. R. Brownyer of the Paige-Detroit Motor Car Co.

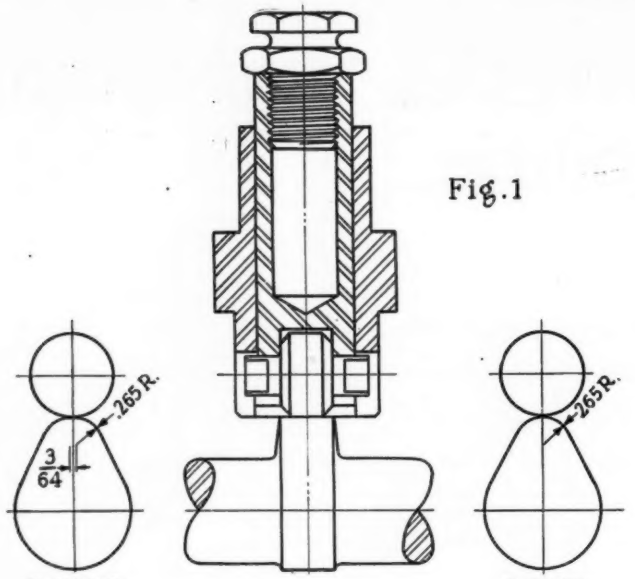


Fig. 1

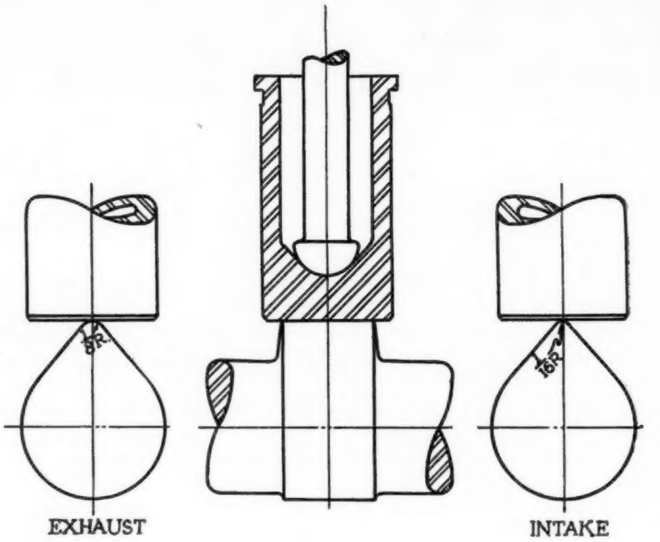


Fig. 2

3. A cylinder rolling and at the same time sliding on another cylinder (Fig. 3) as, for instance, in spur gears, bevel gears, worm gears and helical gears.

For the case of two cylindrical surfaces pressed together by a force W lb. we can determine the width b in. of the rectangular contact surface by the equation

$$\left(\frac{b}{4}\right)^2 = \frac{0.29\,W\left(\frac{1}{e}+\frac{1}{E}\right)}{f\,\left(\frac{1}{r}+\frac{1}{R}\right)} \quad \ldots\ldots\ldots (5)$$

in which

e and E = moduli of elasticity,
r and R = radii in in.,
f = length of contact surface in in.

For calculating the greatest compressive stress C in lb. per sq. in. we have

$$C^2 = \frac{0.35\,W\left(\frac{1}{r}+\frac{1}{R}\right)}{f\,\left(\frac{1}{e}+\frac{1}{E}\right)} \quad \ldots\ldots\ldots (6)$$

For cylinders with the same moduli of elasticity this formula (6) changes into

$$C^2 = \frac{0.175\,W\,E}{f}\left(\frac{1}{r}+\frac{1}{R}\right) \quad \ldots\ldots\ldots (7)$$

Finally, for the case of a cylinder and a flat plate we obtain

$$C^2 = \frac{0.35\,W}{f\,r\left(\frac{1}{e}+\frac{1}{E}\right)} \quad \ldots\ldots\ldots (8)$$

and if $e = E$ we have:

$$C^2 = \frac{0.175\,W\,\mathrm{E}}{f\,r} \quad \ldots\ldots\ldots\ldots\ldots (9)$$

On the basis of the above calculations we can now determine the cam and cam follower dimensions from the wear point of view. Checking up a number of low speed, heavy duty truck and tractor engines, the author found that the greatest compressive stress permissible for both mushroom type and roller cam follower is 80,000 lb. per sq. in., while for high speed engines it amounts to around 60,000 lb.

Example 1—Cam Width Calculation

Find the face of roller cam follower as per Fig. 1, the maximum spring compression load W being 65 lb., radius of roller $r = 7/16$ in., radius of cam R (smallest) $= 1/4$ in.

We calculate the necessary face of the roller by means of equation (7)

$$f = \frac{0.175\,W\,E}{C^2}\left(\frac{1}{r}+\frac{1}{R}\right) \quad \ldots\ldots\ldots (10)$$

$$f = \frac{0.175 \times 65 \times 3 \times 10^7}{(8 \times 10^4)^2}\left(4 + \frac{16}{7}\right) = \frac{0.175 \times 65 \times 3}{64 \times 10} \times \frac{44}{7} = 0.325 \quad \text{in round figures, } 3/8 \text{ in.}$$

Example 2

Find the width of cam for a mushroom type of cam follower, as per Fig. 2, when spring pressure $W = 65$ lb. and the radius of the intake cam $R = 1/16$ in.

Proceeding according to equation (9) we obtain for the face of the cam

$$f = \frac{0.175 \times W\,E}{C^2\,r} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

$$= \frac{0.175 \times 65 \times 3 \times 10^7}{64 \times 10^8} \times 16$$

$= 0.852$, or in round figures, $7/8$ in.

Gear Teeth Sizes

In designing gears we must at the outset make a distinction between two basic conditions of operation, as follows:

1. Where the gears are to transmit nearly the maximum tangential force most of the time; in such cases the tooth size must be determined on the basis of resistance to wear (this applies to tractor gears);

2. Where the gears transmit the full torque only occasionally, in which case the tooth size must be calculated on the basis of strength necessary to insure against fracture when at full load (an example is found in passenger car gears).

There is no hard and fast division line between the two classes.

We may assume that the average loads for passenger car, truck and tractor gears normally are about 20, 50

and 85 per cent of their respective maximum loads. Thus, while it is sufficient to design passenger car gears on the basis of strength (the unit bending stresses must not exceed a given maximum), it is necessary to calculate tractor gears on the basis of resistance to wear of the tooth faces (the greatest compressive stresses on the tooth face must not exceed the critical point). As to the motor truck gears, they are being designed the same way as passenger car gears, the only difference being that the unit stresses are taken considerably lower. However, when it comes to the question of how much lower these allowable unit stresses ought to be, the only dependable guide is to check the compressive stresses on the tooth faces. Tractor gears, too, are being calculated on the basis of bending strength, only that the unit stresses here are taken still lower than for motor trucks. This calculation, however, is of doubtful value, and the only way to ascertain the durability of a gear is to try it out.

There being no safe rule for calculating the capacity of a gear from the standpoint of durability, the designer is compelled to resort to guess work. Tractor gears rarely fail through fracture, the unit bending stresses being low, and, therefore, safe, but they usually wear out on the tooth faces. Apparently the low bending stresses are no safeguard against excessive wear; quite frequently a few gears of a set wear out, while the rest stands up well, yet all were calculated to work at the same bending stress. This, however, can be explained very simply. A coarse pitch does not help to resist the crushing of material on the tooth surfaces. The diameter and face of the gear combined determine the area of tooth contact, while the pitch should be only just sufficient to resist fracture. Any larger pitch than this is uneconomical, there being no increase in power transmitting capacity.

Before we proceed to the solution of the problem of surface stresses, let us first review the method of calculating the strength of gear teeth by the Lewis formula:

$$W = S\ p\ f\ y \text{ lb.} \qquad \ldots\ldots\ldots (12)$$

$$S = S_s \frac{600}{600 + V} \text{ lb. per sq. in.} \qquad \ldots\ldots\ldots (13)$$

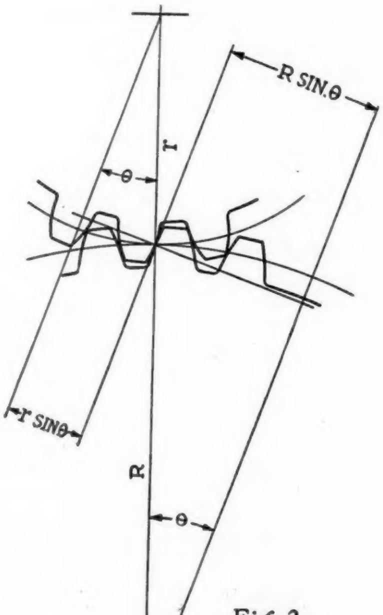


Fig.3

$$HP = \frac{W\ V}{33{,}000} \text{ in.-lb.} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (14)$$

where:

W = maximum safe tangential load in lb. at the pitch circle,

S = allowable unit stress at the given circumferential speed,

p = circular pitch in in.,

f = face of teeth in in.

S_s = allowable unit stress for a circumferential speed of zero, in lb. per sq. in.,

V = circumferential speed in ft. per min.

For the factor y, which depends upon the number of teeth in the gear and upon the pressure angle, we have:

$$y = 0.154 - \frac{0.912}{n} \text{ for 20 deg. obliquity} \ldots\ldots\ldots\ldots (15)$$

$$y = 0.124 - \frac{0.684}{n} \text{ for 15 deg. obliquity} \ldots\ldots\ldots\ldots (16)$$

where n = number of teeth in gear.

As to equation (13) it may be said that it holds well for general machine practice, where rather heavy and large gears and the more common materials, such as cast iron and cast steel, are being used. However, when it comes to heat treated alloy steels, high velocities, accurate workmanship and small, light weight gears, the formula is not so serviceable. Gears for extremely high speeds, when calculated according to this equation, would not be strong enough, while gears for moderate speeds would be unnecessarily large.

Equations (15) and (16) give the values of the coefficient y for full size teeth. For the Fellows stub tooth system these values are different for each pitch, because of the varying ratio of the addendum to the circular pitch. Hence, the strength of the Fellows stub tooth cannot be computed by the Lewis formula unless the pitch is known.

As the pinions usually have only a small number of teeth it will be sufficient to give below the values of y for from 12 to 22 teeth inclusive:

TABLE I

Values of y in Lewis Formula for Different Tooth Systems:

Number of Teeth	15 deg. involute	20 deg. involute	Fellows Stub Tooth					Nuttall Stub
			4/5	5/7	6/8	8/10	10/12	
12	0.067	0.078	0.096	0.111	0.102	0.096	0.093	0.099
13	0.070	0.083	0.101	0.115	0.107	0.101	0.098	0.103
14	0.072	0.088	0.105	0.119	0.112	0.106	0.102	0.108
15	0.075	0.092	0.108	0.123	0.115	0.110	0.105	0.111
16	0.077	0.094	0.111	0.126	0.119	0.113	0.109	0.115
17	0.080	0.096	0.114	0.129	0.122	0.116	0.111	0.117
18	0.083	0.098	0.117	0.131	0.124	0.119	0.114	0.120
19	0.087	0.100	0.119	0.133	0.127	0.122	0.116	0.123
20	0.090	0.102	0.121	0.135	0.129	0.124	0.118	0.125
21	0.092	0.104	0.123	0.137	0.131	0.126	0.120	0.127
22	0.093	0.105	0.125	0.139	0.133	0.128	0.122	0.128
Rack	0.124	0.154	0.173	0.184	0.179	0.172	0.168	0.175

From this table it may be seen that the stub tooth is about 25 per cent stronger than the standard 20 degree involute tooth. Thus we can write approximately

$$y = 1.25\,(0.154 - \frac{0.912}{n} \text{ for Fellows stub tooth.} \,(17)$$

The Fellows system, though working well and widely used in practice, is one of those unfortunate standards which render calculations more difficult. A practical designer must be provided with simple methods of figuring,

otherwise he will not calculate at all but resort to guess work.

Gear sizes for self-propelled vehicles are determined by the following factors:

1. The horsepower, HP, which depends upon the weight and capacity of the vehicle;

2. The revolutions per minute N of the pinion, which depends on the engine speed;

3. The ratio b of the face of the gear to the circular pitch—

$$\frac{f}{p} = b \qquad (18)$$

(This ratio usually lies between the limits 1¼ and 4)

4. The smallest number of teeth n in the pinion, which is determined by interference and the arc of action. From the interference point of view, n varies with the gear ratio g as follows for 20 deg. involute and full size teeth—

g	1	2	3	4	6
n	12	14	15	16	17

In order to avoid interference with a rack, n should not be smaller than its value derived from the following equation

$$\frac{n-2}{2} tg^2 \Theta = 1 \qquad (19)$$

where Θ is the angle of obliquity.

Θ	15°	20°	22°	30°
n	30	18	15	9

To insure continuous and smooth running of the gears the actual arc of action ought to be equal to at least 1¼ circular pitch. In order to accomplish this, a pinion with $\Theta = 20$ deg. must have at least 12 teeth for low speeds, while for higher speeds 15 to 17 teeth are preferable, as we can then obtain more teeth in mesh, and smoother and quieter action.

On the basis of the foregoing consideration we can calculate the size of the pinion (pitch, diameter and face) which would transmit the given load and resist shocks without fracture. The problem usually presents itself as follows: There are given the horsepower HP, the speed N of the pinion and the material, which in turn determines the allowable working stress S.

Now we can decide on the number of teeth n, the coefficient y, the face-to-pitch ratio b, depending on the speed of the gears, accuracy of workmanship, whether the power transmitted is constant or variable and whether the gears are subject to severe shocks. Then we can proceed to the calculation of pitch p and finally of the diameter and the face. This calculation of the pitch is, however, not a simple matter, as it is necessary to use the trial-and-error method and repeat the calculation. This is tedious, but the best plan in the end. Several approximate and simplified formulæ have been devised, in order to get a closer and quicker approximation for preliminary work. But since these methods cannot take all the factors into consideration, corrections have to be made, and the result is correct only within limits. This means repeated calculation and slow progress.

There is, however, a method of obtaining exact results for any given conditions, by the use of the ordinary slide rule and the chart in Fig. 6. In order to demonstrate this method, let us begin with equation (12) and substitute gradually for all unknown quantities until we obtain the pitch as a function of the known factors:

$$p = \text{function of } HP, N, S_s, b, n, y$$

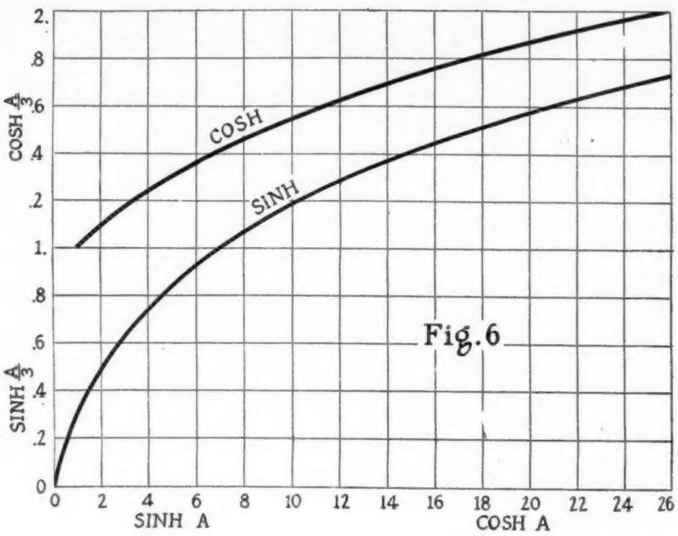


Fig. 6

We know that

$$W = S\,p\,f\,y \text{ lb.}$$

$$f = b\,p \text{ in.}$$

$$S = S_s \frac{1}{1 + \frac{V}{600}} \text{ lb. per sq. in.}$$

$$V = \frac{n\,p}{12} N \text{ ft. per min.}$$

Hence,

$$W = S_s\,b\,p^2\,y \frac{1}{1 + \frac{n\,p}{12 \times 600} N}$$

Then according to equation (14)—

$$HP \times 33000 \times 12 \left(1 + \frac{n\,p\,N}{12 \times 600}\right) = S_s\,b\,p^3\,y\,n\,N,$$

from which we obtain

$$p^3 - \frac{55\,H\,P}{S_s\,b\,y} p - \frac{396000\ HP}{S_s\,b\,y\,n\,N} = 0$$

This equation can be easily solved by means of the hyperbolic cosine of an auxiliary angle A. In fact, it is not necessary for the reader to be acquainted with these functions at all, as the required values can be readily found from Fig. 6.

We have for the auxiliary angle A:

$$cosh\,A = \frac{2.52 \times 10^3}{n\,N} \sqrt{\frac{S_s\,b\,y}{HP}} \qquad (20)$$

and for the pitch

$$p = 8.56 \sqrt{\frac{HP}{S_s\,b\,y}} \cosh \frac{A}{3} \text{ in} \qquad (21)$$

To simplify the calculation I have constructed a curve (Fig. 6) in which values of cosh $\frac{A}{3}$ are laid off as abscissæ and cosh A as ordinates. For any value of cosh A obtained by means of equation (20) we can find a corresponding value of cosh $\frac{A}{3}$. Knowing the value of cosh $\frac{A}{3}$, we can determine the pitch by means of equation (21).

(To be continued)

Influence of the Magnetic Circuit on Magneto Performance

Part I

As a writer on electricity Mr. Geist is well known to the readers of AUTOMOTIVE INDUSTRIES. In this first installment of his article he gives the fundamental considerations considerable attention and then takes up the operation of the "H" form of magneto in detail. Other types will be studied in the second half of this article, which will be published later.

By Harry F. Geist, E. E.

EVER since the first self-contained high tension magneto made its appearance, it has been almost universally accepted that a high tension magneto, in order to be a practical and serviceable machine, must consist of the combination of a magnetic field, a primary winding, a secondary winding, a breaker and a condenser.

While a great deal of experimental work has been done in an endeavor to reduce the number of elements required, to simplify the construction and reduce the cost of manufacture, the machine still remains a combination of the above five elements.

On the other hand, engineers and inventors, in an endeavor to meet the ignition requirements of the different types and designs of internal combustion engines, ranging from the slow speed, single cylinder, cheap farm engine, to the high speed, multi-cylinder high class airplane engine, have developed a large number of magneto types based upon the use of those five necessary elements.

The principal difference between the different types is in regard to the manner in which the rotor shifts the magnetic flux from the permanent magnets with respect to the windings for current generation. The influence of the necessarily different forms of magnetic circuits employed is largely responsible for the difference in the performance characteristics of these machines.

The first magnetos on the market were of the well known bi-polar, wound armature construction. The departure from this recognized construction has been in most cases an attempt to produce an efficient machine having stationary coils, thus eliminating the use of collector spools, rotating breaker parts, etc., and not especially to gain in electrical efficiency.

Magneto Development

The advantage of stationary coils is regarded by some as even warranting a sacrifice in electrical efficiency.

In some cases the departure from the wound armature type was made with the object of producing more sparks per revolution of the rotor than is possible with that type, so as to make the machine adaptable to twelve and sixteen cylinder engines without excessive rotor speeds.

Other efforts in magneto development have been made for the sole purpose of reducing the cost of manufacture, covering both the items of material and workmanship. These efforts were made principally to satisfy the cheap engine trade, but engine manufacturers in general have now about learned that a magneto cannot be a satisfactory machine unless the best of material and workmanship enter into its construction.

For the purpose of this article, the writer has classified the magnetos on the market into three general types. These are the "Wound Armature" type, the "Rotary Field" type and the "Inductor" type. It is the writer's purpose in this article to discuss machines representing these different types and to point out something of their performance characteristics and of how their behavior is influenced by the magnetic circuit employed. The scope of the treatment may be considered to include anything of interest that influences or affects the performance of a magneto.

Electrical Introduction

Before entering upon a discussion that may necessarily involve some electrical terms, it may be well to first give a brief explanation of the important electrical quantities.

Electricity presents itself in two different forms, depending upon the circuit conditions under which it is generated or utilized. These two forms are called electro-magnetic energy and electro-static energy. The first form manifests itself by the magnetizing influence it exerts, while the latter form reveals itself with a force that tends to break down circuit insulations.

Electrical current is a principal factor in the measurement of electro-magnetic energy, and electro-motive-force is a principal factor of the electro-static form of energy.

Electricity, like all other forms of energy, expends itself in the form of heat, and resistance is the characteristic of the circuit through which this energy loss or transformation takes place. This resistance may be a characteristic of the electrical circuit, or it may be a resistance in the magnetic circuit to eddy currents, etc., but wherever it exists, it absorbs energy.

When the amount of electrical energy supplied to a circuit is greater than that used, it is being stored. In electrical circuits we have inductance as a circuit characteristic that gives rise to the storage of electro-magnetic energy and capacity as another characteristic giving rise to the storage of electro-static energy.

The storing or dissipation of this energy requires time, so that time enters into all phenomena involving the three quantities, resistance, inductance and capacity.

Inductance is a characteristic of special importance in magnetos, and it will be well to say something about it. It is a term that expresses the degree of interlinkage between the turns of a coil and the lines of magnetic flux established by the current flowing in the coil. Such an interlinkage represents stored electro-magnetic energy that may be expressed by the following equation.

$$W_m = \frac{L\,i^2}{2} \quad \text{.....................} (1)$$

in which L designates inductance and i represents current. Note that the value of such energy depends upon the square of the current and only directly upon the inductance.

Coils may be made with or without iron cores, and if iron cores are used they may have widely different forms. The inductance in a coil without an iron core is relatively small and will change only little with the shape of the coil, as compared with the inductance with an iron core. The more completely this iron core is interlinked with the coil the greater will the inductance be.

Briefly, inductance is to the electrical circuit what mass is to a flywheel—it permits the storage of energy and stabilizes its action.

Capacity

Capacity for the electro-static form of energy is due to the proximity of circuit areas that are subject to differences of electro-motive force. It may be concentrated into a small body built especially for that purpose, as in the electrical condenser, or it may exist as distributed capacity within the windings of a coil due to the proximity of turn to turn, layer to layer, or of the coil as a whole to other parts of the machine.

The amount of electro-static energy that will be stored in a circuit is expressed by the following equation.

$$W_c = \frac{C\,e^2}{2} \quad \text{...................} (2)$$

in which C designates the capacity and e represents the electro-motive force or e.m.f. Note in this equation that the amount of energy depends upon the square of the e.m.f. and only directly upon the amount of capacity.

The amount of distributed capacity in a winding is very small as compared with that contained in a condenser, but in cases where very high potentials are employed, as in a magneto secondary winding, this distributed capacity becomes of considerable importance. In the case of a condenser where portions of electrical circuits of considerable area are brought very closely together, the capacity becomes sufficient to store useful amounts of energy at comparatively low potentials.

Time

Science has found that electricity has about the same velocity as light, but this speed is a measure only of the time required for an electrical force to get to a place ready for work. The time required for the electrical energy to do its work is a very different matter and usually is of sufficient magnitude to be readily measured.

In the case of electro-magnetic phenomena, resistance and inductance determine the time required, while for electro-static phenomena the time is determined by the resistance and the capacity. Both forms of energy may be present at the same time and in that case the time required will depend upon all three circuit characteristics.

In general, electro-magnetic phenomena are very much slower than electro-static phenomena, so that when both inductance and capacity are of importance in a circuit the phenomena are quickened through the division of the work. That is, capacity tends to speed up the inductive circuit.

The time-constant of the electro-magnetic circuit is expressed by

$$T = \frac{L}{R} \quad \text{.....................} (3)$$

and that of the electro-static circuit is

$$T = C\,R \quad \text{.......................} (4)$$

Desirable Magneto Performance Characteristics

Before going into the study of magnetos as they are to-day, it may be of interest to consider the ideal magneto toward which engineers, inventors and scientists are working.

From a mechanical standpoint, a good magneto must have sufficient clearance between the rotor and the stator so that the danger of striking under ordinary conditions of usage in service is reduced to a minimum. This clearance should be from 0.010 to 0.012 in. all the way around. But where the distance between the rotor bearings is short this may be reduced to 0.008 in. and possibly less in special cases without danger, but clearances as small as 0.0025 to 0.005 in. are dangerous.

A magneto should spark at as low a speed as is possible, so as to make the starting of an engine easy without the use of a spring starting coupling. The writer has seen magnetos spark across a 3/16 in. standard three-point gap at speeds as low as 45 r.p.m., and has heard reports of cases where speeds as low as 35 r.p.m. yielded sparks. In the cases that have come to the writer's attention, the clearance gaps were very small, and he has come to associate small clearance gaps with reports of unusually low speed performance.

When the standard is set for low speed magneto performance, let us trust that consideration will be given to the clearance and the distance between rotor bearings.

Magneto Design

An internal combustion engine requires its best ignition spark at starting and where the engine is cranked at low speed, the spark must usually be delivered with the spark lever, armature, etc., in the retarded position. It is, therefore, desirable to design the magneto so that it delivers its best spark at the retarded armature position and yet gives a suitable running spark for the advanced armature position.

Another desirable performance characteristic is uniformity; that is to say, above a certain speed of operation the spark energy should not increase materially with an increase of speed.

A good magneto should have sufficient sparking range to take care of at least 30 deg. difference in firing positions. If this firing range is taken care of mechanically, instead of by the inherent performance characteristics of the machine, the latter should at least have sufficient range of energy so that an adjustment of the breaker contact points or other parts does not throw the spark sufficiently off "peak" to materially reduce its firing ability at low speeds. That is, a magneto must have sufficient range so that it is not sensitive as regards the relation between the interruption of the primary circuit and the magnetic break between the rotor and the stator.

Magnetos of the high tension type have secondary windings that comprise anywhere from 8000 to 14,000 turns depending upon the type. It would be a very desirable improvement in magneto design if the functioning of the different elements of a magneto could be so harmonized that a greater voltage could be obtained per turn of the secondary circuit and thus the number of turns materially reduced. A 50 per cent reduction in the number of secondary turns required would be a big improvement and would incidentally make possible a reduction of the primary condenser capacity.

In Fig. 1 are shown the principal features of the magnetic circuit and the windings of an "H" armature type magneto. This magneto delivers two sparks per revolution of the armature and is practical on engines having as many as eight cylinders.

In discussing this type of machine, it might be well to begin with the low tension type, because it was from that machine that the modern self-contained high tension magneto originated.

The wound armature is the only type of magneto that has given satisfactory results as a geared-to-the-engine, rotating, low tension magneto ignition system. The reason is that this machine is the only one that generates sufficient energy and stores it over a sufficient firing range to satisfy the requirements of a low tension ignition system, wherein the magneto must be timed to a separate igniter mechanism and produce a spark for either "advanced" or "retarded" firing positions.

Fig. 1 shows a magnetic field, comprising a pair of "U" shaped magnets with like poles adjacent terminating at a pair of pole shoes. These pole shoes are usually made of a soft cast iron and are so formed as to present a tunnel for a rotating armature between them. A more recent practice, both in England and in the United States, is to build these pole shoes of sheet steel laminæ. The polar embrace is about 90 deg. and the poles should be diametrically opposite each other.

The armature core consists of a number of soft sheet steel laminæ of "H" form clamped by screws or rivets between soft cast iron end pieces which are so shaped as to form continuations of the armature faces of the laminated portion and at the same time to form a continuation of the recess for the windings at the ends of the armature.

The Armature Faces

The coil is wound in this recess, the first turn usually being grounded to the core, but otherwise well insulated.

The armature faces must also be diametrically opposite each other, so that symmetrical co-action will always take place between the two pole shoes and the two armature faces, and these faces must have sufficient width so that when the armature is in the vertical position, as shown in Fig. 1, they span the gap between the opposite pole shoes and in addition "over-lap" slightly with the pole tips. This over-lap insures that the armature will always serve as a "keeper" to protect the magnets from the demagnetizing effects of an open circuit.

The depth of the armature faces along the bore is equal to at least the total over-all length of the winding. This depth also determines the depth of the pole shoes along the bore and is known as the "active magnetic depth" of the machine. The "active winding depth" is only approximately one third of that amount, including as it does only the depth to which the laminæ are stacked plus a small tie between the opposite end wings at both ends of the armature.

From an electrical standpoint, the most important feature in the design of a magneto is its magnetic circuit, as represented by the magnets, the armature and the pole shoes, and it is in this respect particularly that the different types of magnetos vary.

The magnetic circuit performs three important functions in a machine. First—it conveys the magnetic flux from the permanent magnets into interlinkage with the winding. Second—it provides a means for shifting or reversing the flux with respect to the winding. Third—it provides a circuit or circuits for the reactive flux so as to give the machine energy storage ability.

In the wound armature machine, illustrated in Fig. 1, the magnetic path formed by the pole shoes and the arma-

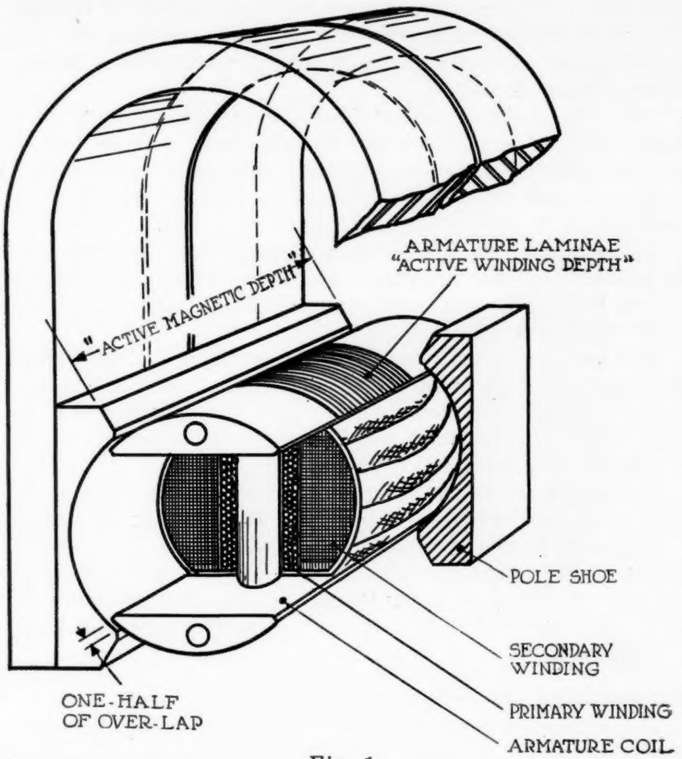


Fig. 1

ture core faces, when the armature is in the horizontal position, is very short, and, therefore, affords a very effective means for interlinking the flux with the windings.

As the armature advances in its rotation toward the vertical position shown, there is a slight tendency for some of the flux to pass out of interlinkage with the coil, but the flux for the most part will be drawn toward the tips of the separating pole shoes and armature faces, increasing in density as the area of co-action decreases.

When the armature reaches the vertical position the faces themselves span the polar gap and form a by-pass magnetic circuit that takes all the flux from interlinkage with the coil. But this perfect by-pass condition is only momentary in the rotation of the armature and as the armature advances the flux begins to rapidly thread back through the coil in the opposite direction.

The most important feature of this reversal of flux with respect to the coil is that most of the flux reverses during the period of a relatively small angle of motion in the neighborhood of the vertical position, and the tendency therefore is to produce a strong impulse of energy that is of sufficient intensity to be of value for ignition work. The secret of conditions that tend to produce a very quick change of flux direction with respect to the coil lies in the long "active magnetic depth" of the machine and the short "over-lap" which make a long and narrow over-lap area for the position of maximum flux change. These features are indicated in Fig. 1.

But during the period of heavy flux change the winding being short circuited and current flowing, armature reaction and inductance effects enter into play in a very pronounced manner and change the tendency for a very "peaked" wave of generated energy to a more distributed wave by exerting a sort of retarding action on the flux shift. The resulting wave has a high energy storage value over a range of from 30 to 40 deg. of motion and is capable of delivering a useful spark over that range.

The impulse or wave of energy, it is seen, occurs just after the armature passes the vertical position. A study of the magnetic circuit as interlinked with the coils for the neighborhood of this position will show that the two

pole shoes form an almost complete iron circuit in connection with the armature core for the coils, so that the coil is in reality almost completely surrounded with a highly magnetizable material.

This ironcladding effect tends to give the coil a very high value of inductance per turn, as was pointed out in the introductory paragraphs on Inductance. The inductance per turn of the coil is about as high in this class of machine as it is possible to obtain.

Along with the high value of inductance, it must also be evident that the resistance per turn of the coil is about as low as it can be in such a machine, due to the relatively short "active winding depth."

The low resistance, representing low energy loss in the coil, and the high inductance, meaning a high ability for energy storage, coupled together give the machine a relatively high time-constant which is responsible for the range of energy storage. See equation (3).

Furthermore, as the magneto is speeded up, this same time-constant prevents the energy generated from responding directly to the increased rate of flux shift, so that the result is an added energy storage range rather than increased intensity at any instant. In this manner the machine is rendered uniform in its sparking behavior at different speeds.

Summarizing the Influence

The importance of the iron cladding effect of the pole shoes is, therefore, seen to reveal itself in both the firing range and the sparking uniformity of the magneto.

Summarizing the influence of the magnetic circuit on the generative and storage ability of the low tension armature wound magneto, it is seen that the shortness of the magnetic circuit insures a maximum amount of flux interlinked with the coil, reducing the leakage or stray flux to a minimum. It has a long and narrow over-lap area that tends to give a quick and decided reversal to the flux even at low speeds. By "reversing" the flux with respect to the coil it is seen that the flux from the magnets is used twice upon the coil in producing one impulse or wave of energy. The coil has a low resistance per turn and a relatively high value of inductance per turn. These are properties that promote efficiency, high storage ability, range and uniformity of sparking action.

Besides the above it will be noticed that there is a perfect symmetry in the design of the magnetic circuit of the wound armature type magneto, so that the phenomena taking place on one side of the coil are identical at all times with those occurring on the opposite side.

It is from such a machine that the self-contained high tension magneto had its origin and it is for the same reason that the high tension type is a very efficient and satisfactory ignition device, retaining as it does all the inherent characteristics of the low tension machine and in addition delivering a high tension spark that can be readily distributed to the different cylinders of an engine.

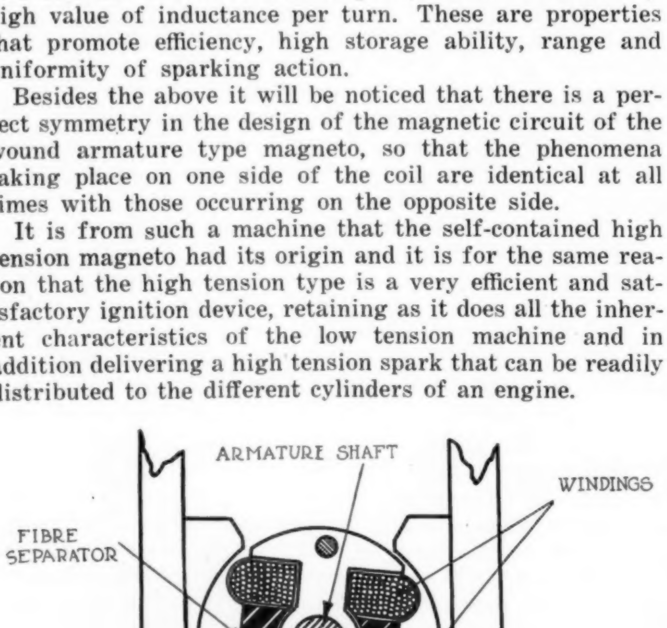


Fig. 2

The magneto with its permanent magnets is what is known as a constant field generator. Under the condition of a constant excitation field, the amount of energy that will be generated in the winding is within very wide limits practically independent of the number of turns and the space taken up by them.

Because of this fact, it is possible to reduce the space occupied by the generating winding to a fraction of the total winding space and give the armature a secondary winding. Such a machine, having a mechanical interrupter and a condenser for its generating or primary winding, and having a secondary coil of many turns of fine wire, becomes what is known as a self-contained high tension magneto, and performs all the functions of a high-tension ignition system in a single structure.

Besides having all the meritorious characteristics attributed to the low tension magneto from which it sprang, it delivers its energy through a transformation in the form of a high tension or jump spark.

Under the influence of the high voltage that is induced in the secondary circuit following the interruption of the primary circuit, the distributed capacity that exists between the turns and layers of the secondary coil and between the coil as a whole and the neighboring parts of the machine, becomes of considerable importance, so that besides the inductance and the resistance that this secondary circuit has, its performance characteristics must also be governed by the amount of distributed capacity present. The effect of this will be discussed later.

Experimental and Development Work

A great deal of experimental and development work has been done in an endeavor to get an armature wound magneto that will spark efficiently at low speeds and at retarded armature positions. A great many different forms of pole shoes have been designed. For the most part they consist of a compound polar tip for co-action with the trailing tip of the armature face during rotation, so that a partial magnetic shift is produced during the advanced armature position that is followed about 30 deg. later by a completion of the flux reversal. The idea involved is to have some mechanical flux shift taking place for the retarded position rather than to depend entirely upon the storage of energy. This subject of special pole shoes is taken up in an interesting manner by Fred J. Hoffman in an article on pole shoes in AUTOMOTIVE INDUSTRIES of April 18, 1918, and the writer is referring to that article as of special interest in connection with armature wound magnetos.

The wound "H" armature type magneto represented in Fig. 1 journals on two shaft extensions that are concentrically assembled to armature core, so that this machine does not have the mechanical advantage of a solid shaft.

Some armature wound magnetos having solid shafts have been marketed and give good results, but they do not have the same desirable performance characteristics that are found in the wound "H" armature type. As far as the writer knows, this solid shaft type has never been produced in any other than the low tension models.

Fig. 2 is intended to illustrate the essential features of this "solid shaft" type of magneto as differing from the usual "H" form. In this machine it is necessary to divide the winding into two separated sections, using a fiber spacer block to support the inner sides of the coils.

Another structural difference between this form of magneto and the "H" type is that the armature core is usually composed only of laminæ riveted together and pressed upon a shaft. In order to build up the "active magnetic depth" to a suitable length, the "active winding depth," which in this type of construction is equal to it, builds up

correspondingly and results in an increase in the amount of resistance per turn of the coil.

In addition to this, with the winding separated into two sections, there is a decrease in the circuit inductance. With an increased resistance and a decreased inductance, the range and the uniformity of sparking are not so good as in the "H" type. For the range that it does have, however, it will give as good a spark as the "H" type, and if caught on its "peak," a better one.

The solid shaft feature is a decided mechanical advantage for the class of service that this particular type of magneto is called upon to perform.

Rotating Field Magneto

Fig. 3 is presented to show the principal features of the magnetic circuit of a rotating field type magneto, and represents this type as successfully manufactured in both the United States and England.

This machine was gotten up primarily for the purpose of producing a high tension magneto having both stationary coils and interrupter parts. It can be made in types giving two, four, six or more sparks per revolution and is therefore practical on engines having twelve or sixteen cylinders without excessive speeds.

The magnetic circuit comprises a pair of "U" shaped magnets with like poles adjacent, that terminate at a pair of iron bearing plates. The rotor which journals in these bearing plates consists of two iron lugs, magnetically separated by a bronze block, to which they are securely riveted by brass pins. Each rotor lug has a few thousandths of an inch running contact with its bearing plate, so that it is always maintained at a constant magnetic polarity by the magnet pole with which it is connected, and the rotor constitutes, virtually, a rotating magnetic field.

Co-acting with these rotor lugs are two laminated core extensions that form a tunnel for the rotor. These core extensions are located midway between the span of the magnets and sufficiently distant from each pole of the magnets so that magnetic leakage is kept low.

For the two spark magneto illustrated in Fig. 3, each of the laminated core extensions has a polar embrace of about 90 deg., while the rotor lugs are given sufficient width so that they over-lap the gap between their tips.

The coils, consisting of a primary and a secondary winding, are wound directly upon a separable laminated core that fits across the upper extremities of the core extensions, completing the magnetic circuit with the rotor lugs. The coils are therefore mounted up in the arch of the magnets.

When the rotor is in the so-called horizontal position, the magnetic flux passes from the north pole of the magnets, through the bearing plate to its rotor lug, and thence up the adjacent core extension to the coil core, through the coil and back through the other half of the circuit to the south pole of the magnet. For the vertical position, the flux is by-passed from the coils by the 90 deg. arcs of the laminated core extensions.

Thus, as the rotor revolves, it reverses the flux with respect to the windings and tends to generate a wave of energy in the same manner as shown for the wound armature type magneto. This wave of energy is produced just after the rotor passes the vertical position.

It will be seen from the illustration that this design of magneto has a considerably longer magnetic path through the coil core connecting the two rotor lugs, than the path connecting the pole shoes of the armature-wound machine. This long magnetic circuit, together with the closer arrangement of iron parts subject to magnetic forces, causes a considerable magnetic leakage. There is, furthermore, an additional air-gap between the bearings

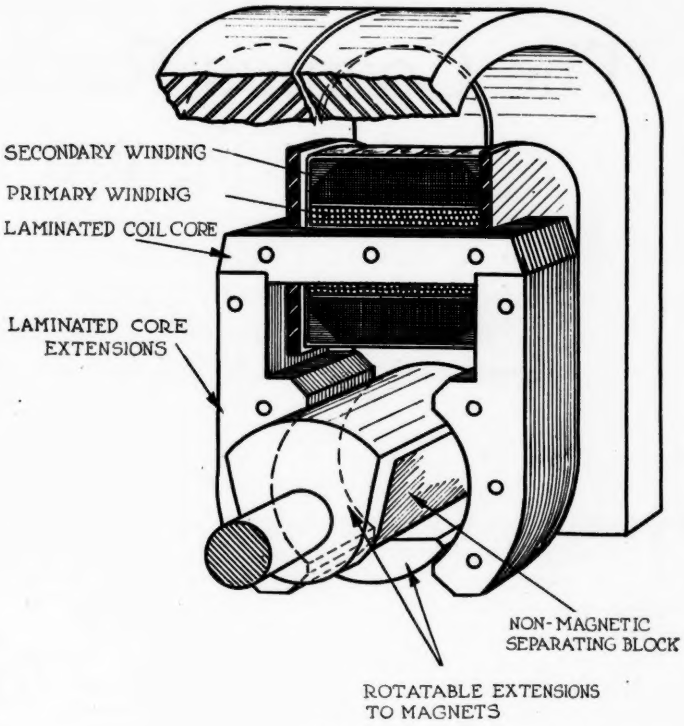


Fig. 3

and the rotor lugs that affects the amount of magnetic flux.

The result of this reduction in the flux passing through the coil core is that the generative ability of the machine is not as great as that of the wound armature type.

A further point worthy of note in connection with the generative ability of the machine is that the "active magnetic depth" of the machine is very short, being exactly equal to the "active winding depth" of the machine. Owing to this shortness of "active magnetic depth" the over-lap required is a little greater, so that the same speed of operation in this type of machine does not tend to produce as quick a shift of magnetic flux. However, for reasons that will be shown presently, the coils in this type of machine do not retard the tendency for a quick shift in as pronounced a manner as those in the other machine, so that this type seems to respond more naturally to the action of the rotor.

(To be continued)

A New Method of Dynamically Balancing Crankshafts

AT a recent meeting of the S. A. E. Minneapolis Branch, C. W. Pendock of the Le Roi Engine Co. described a new method of balancing engine crankshafts. After having secured a fairly good static balance the shaft is suspended by a woven cable of hemp or cotton and is then rotated at speed. The shaft is hung from the cable by means of a ball and socket, the ball being about 1 in. in diameter and having a hole drilled through it through which the cable is threaded.

The shaft being rotated by means of a variable speed motor, is brought up to speed gradually, and if out of balance it will begin to quiver. At a certain speed the shaft will shake quite violently. It is marked by means of a marker in the usual way, and the point on the shaft where the mark occurs is not the heavy side as might be supposed, but the light side. Mr. Pendock is not very clear in his explanation of why this should be so, but experience has confirmed the observation.

The Cosmos Radial Airplane Engine Designed in England

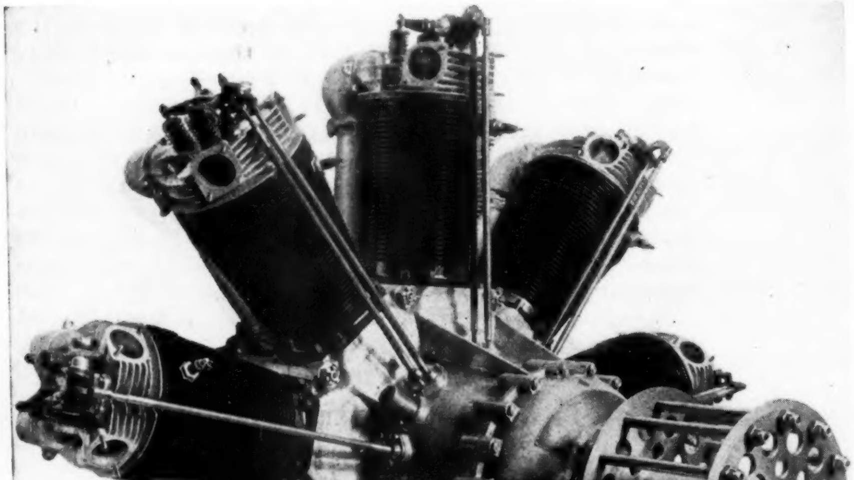
In the Statistical Issue of AUTOMOTIVE INDUSTRIES, in reviewing the trend of American airplane design, it was said that "designers would welcome a radial engine similar to the Salmson and Cosmos types of foreign practice." This article describes the Cosmos, a British radial engine.

By M. W. Bourdon*

THERE is hardly any question that if the war had been prolonged one of the most favored of static air-cooled radial engines for the airplanes of the British Air Service would have been the Cosmos, designed and made by the Cosmos Engineering Co., Ltd.

The type was originally put in hand to meet certain specific requirements as to overall dimensions, and the first model behaved so well that other sizes were developed from it without the same limitations being imposed. At the time of the armistice one of these models was going into production, but the latter was cut off short by the British Air Ministry, though it is stated that the Ministry have no intention of allowing the design to be discarded but will adopt it as a standard when the post-war program is made up.

Mr. Bourdon is the engineering correspondent of AUTOMOTIVE INDUSTRIES *in the United Kingdom.*



Front end of the Jupiter Cosmos engine

The four sizes designed up to the present are as follows:

"Lucifer," 100 hp. three cylinders, one row, 5¾ x 6¼ in.

"Mercury," 300 hp. fourteen cylinders, two rows, 4⅜ x 5 13/16 in.

"Jupiter," 450 hp. nine cylinders, one row, 5¾ x 7½ in.

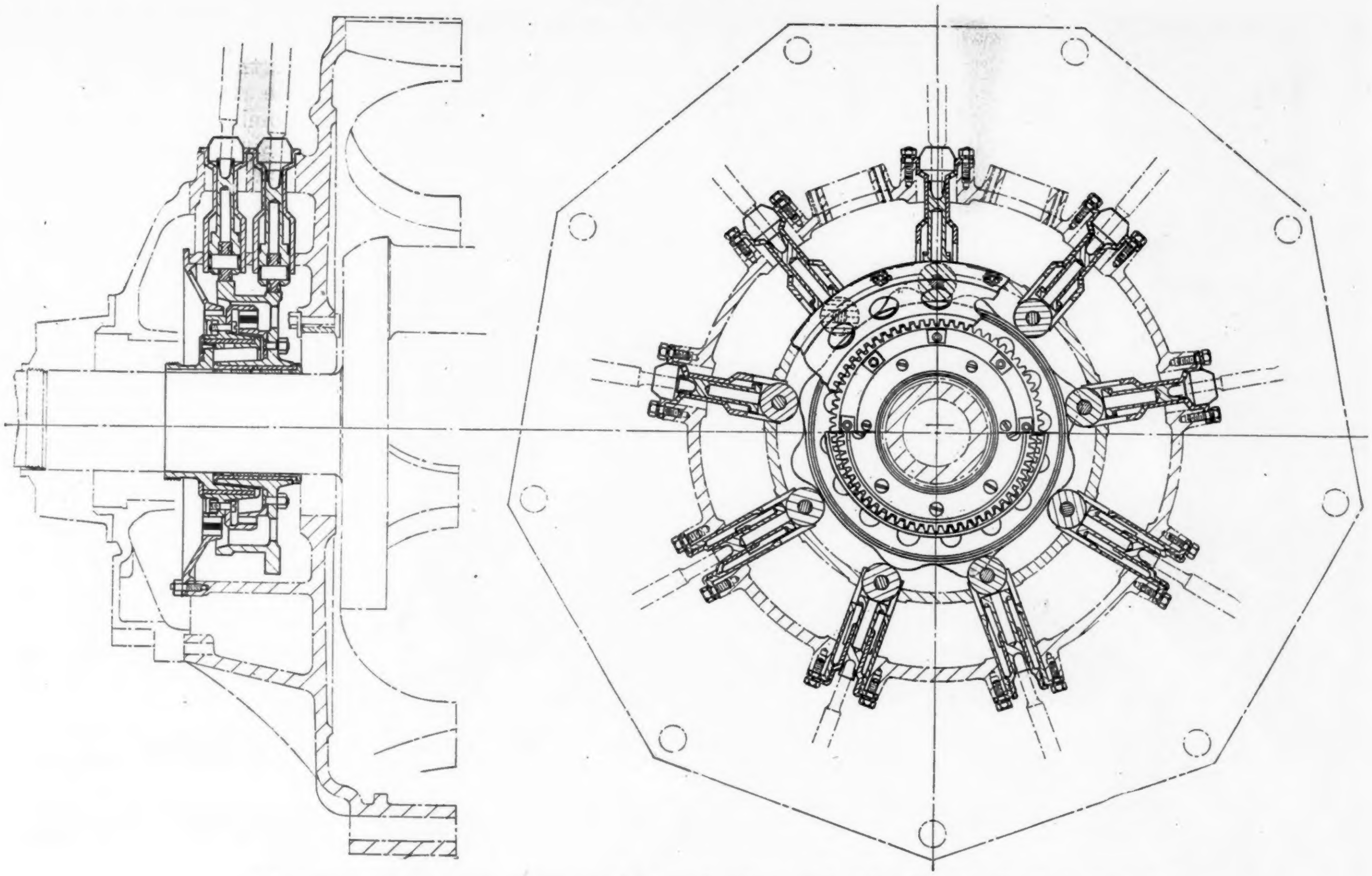
"Hercules," 1,000 hp. eighteen cylinders, two rows, 6¼ x 7½ in.

Examples of all except the largest have been made and tested. Some are still flying, after considerably more than 12 months from the time of being installed.

The size in which the British authorities had and still have most interest is the Jupiter. Its outstanding features are (1) low weight per b.h.p., (2) suitability for production, (3) potential durability and (4) accessibility and detachability of units.

Except in regard to detail, the four designs are practically identical, although in the Mercury—the original type—there are only three overhead valves, two exhaust and one inlet, as compared with four in the others. The Mercury, it is of interest to note, under test in April, 1919, by the Air Ministry in a Bristol Scout F., was responsible for what were at that time two record climbs; the machine ascended to 10,000 ft. in 5 min. 25 sec., and to 20,000 ft. in 16 min. 15 sec., the speed at the lower altitude being 143 m.p.h.

The best idea of the general construction of all four types is obtained by a study of the Jupiter, which not only embodies all the special features of the series but was the engine for which production plans were most advanced at the end of hostilities. This plant was to be used in Bristol, Sopwith and Westland fighters. It has its nine cylinders arranged in a single row; each cylinder with its radiating fins of 0.3 in. pitch is machined from the solid steel forging with an integral crown ¾ in. thick, in which are the seats of the four valves, but with a separate aluminum head carrying the phosphor bronze valve guides.



Arrangement of cam sleeve, rocker mechanism and epicyclic distribution gear

The inlet valves are operated by push rods and Y shaped rockers, the single arm of each of the latter being cupped to receive the top end of the push rod, while the forked ends have adjustable contact pins applying to the valve stems. A similar push rod for the exhaust valves actuates a pair of rockers but applies to a cup at the center of a link piece between the two outer ends. The distribution gear, inclosed within the crankcase extension at the front—the engine is designed for a tractor propeller—consists of a train of two pairs of eccentric and concentric internal and external pinions rotating the cam drum at one-sixth engine speed, there being three inlet and three exhaust cams applying successively to the roller-ended tappets. In effect, the distribution is an epicyclic system, the reaction from the torque being taken by a stationary and internally toothed ring bolted to the cylindrical extension of the crankcase which incloses the gearing and cam drum.

Tulip type, concave headed valves are used, the stems being drilled through more than half their length from the threaded outer ends. Each valve has two concentric helical springs—the outer one being rectangular in section—retained by a plain bored cone and a D washer having two vertically projecting tongues that engage with slots in the under face of a lock nut. To remove the springs and valve, the washer is pushed along the stem against spring pressure, thus clearing the nut and allowing the latter to be unscrewed.

Special provision is made for lubricating the valve rocker spindles. The latter, of large diameter and hollow, are made to form oil reservoirs; they have capped ends and with an aluminum unit inside that divides up the interior into three oil compartments, one for each of the three rocker bushes. The oil fed therein by hand serves for many hours' running.

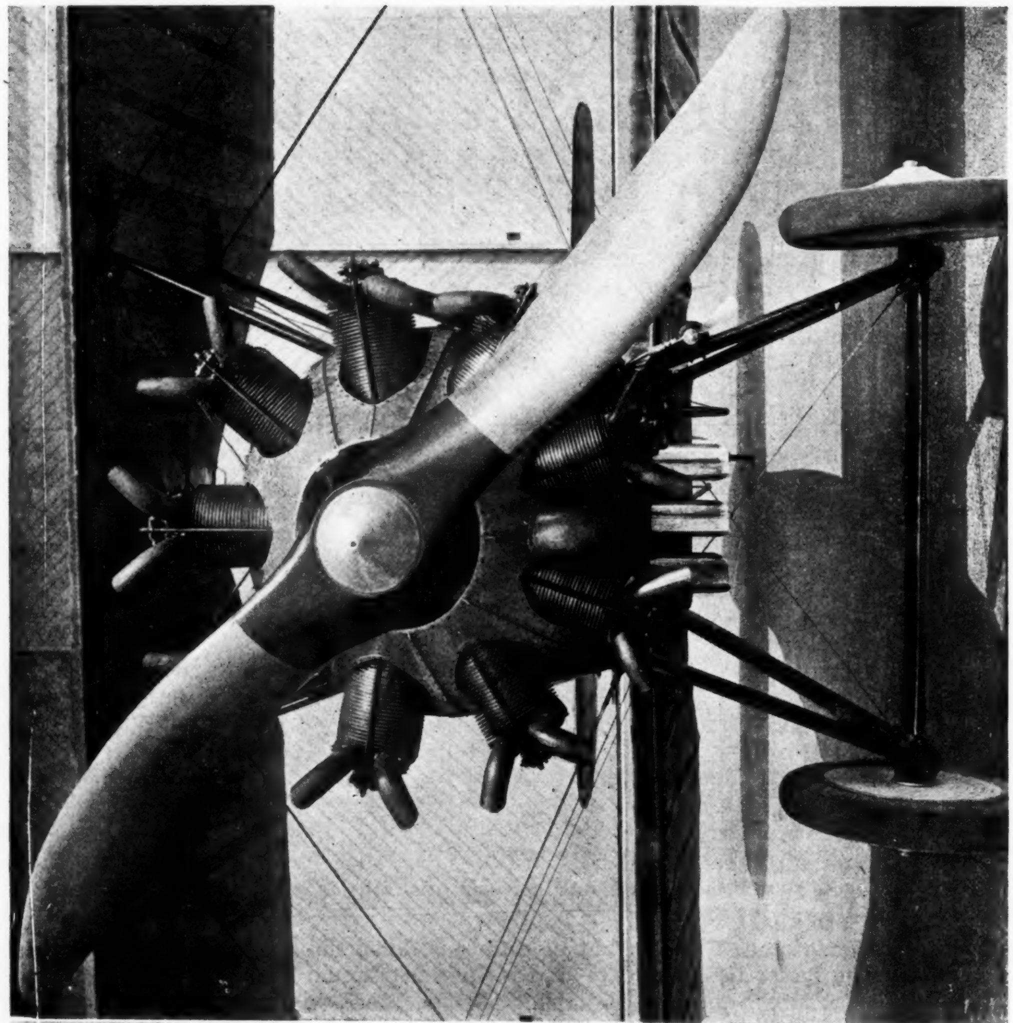
The valves for inlet and exhaust are identical in design but of different dimensions and material. The inlets are of the British standard specification known as K.9, a nickel chrome steel, while the exhausts are a tungsten alloy termed K.8; the effective diameter of the inlets is 1 7/8 in., that of the exhaust being 1 25/32 in.

Pistons are of a modified slipper type in aluminum, with three rings in the crown and a scraper groove below them. The piston pin floats in both piston bosses and small end, being prevented from moving axially by an integral flange at one extremity and a cap held by stud and nut to the closed end of the pin at the other side.

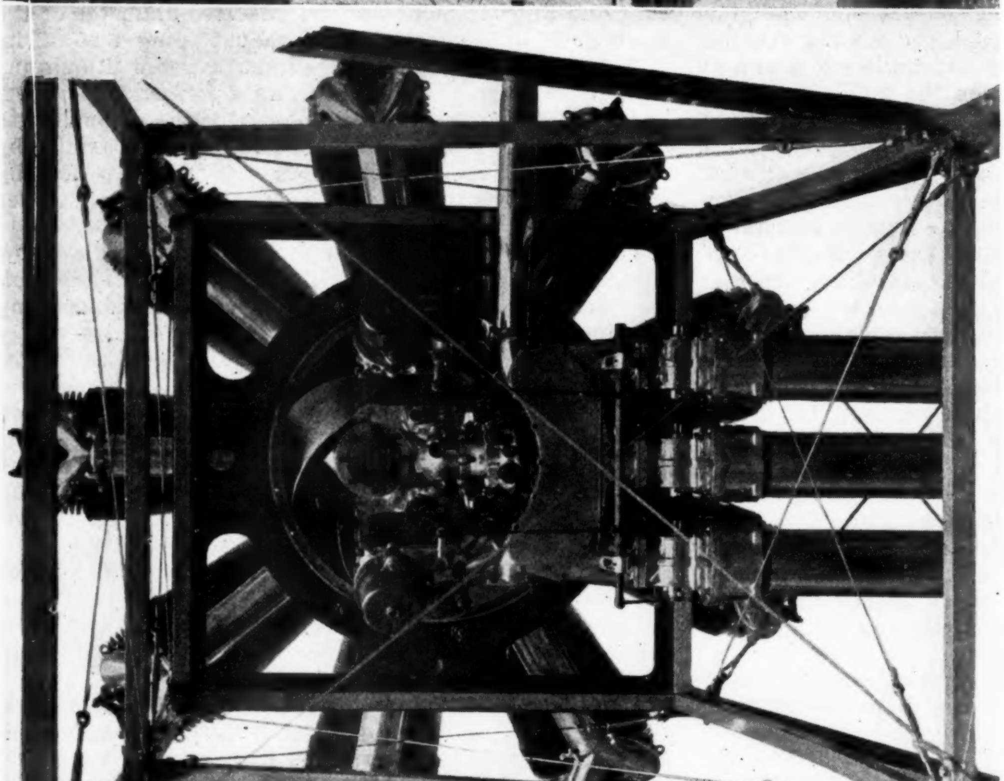
The master connecting rod applies to the single-throw counter-balanced crankshaft through a white metal lining, parallel extensions at each side of big-end and cap forming the bearing supports of the bob weights—a special feature referred to later in detail. Of the eight articulated rods, four are anchored to the rod end and four to the cap, which is held up by two bolts on each side.

A phosphor bronze bush is fitted at each end of the articulated rods, the lower ends of which are secured by hollow pins prevented from moving axially by a tapered abutment at one end, and a cap secured by a nut and bolt passing through the center. These pins are not bored right through, but have an interior web closing off two-thirds of the length to form an oil reservoir, lubricant being fed thereto under pressure from the main big-end bearing.

A dry sump system of lubrication has, of course, been adopted. The scavenging and pressure pumps, of the gear wheel type, are carried as a unit in a detachable aluminum casing bolted to the back of the crankcase and run at half engine speed. A small bevel pinion secured to the crankshaft extremity drives a larger bevel on the vertical pump shaft, which carries the pinions of



Jupiter engine mounted in Bristol Badger airplane



Rear view of Jupiter engine, with cover of mixture distributer removed

both scavenging and pressure pumps, the capacity of the former being double that of the latter. Combined with the pump casing is the pressure relief valve, while the summit of the pump shaft bears a helical pinion for the tachometer drive. A cylindrical strainer intercepts the oil passing to the scavenging pump and another as it leaves the pressure pump.

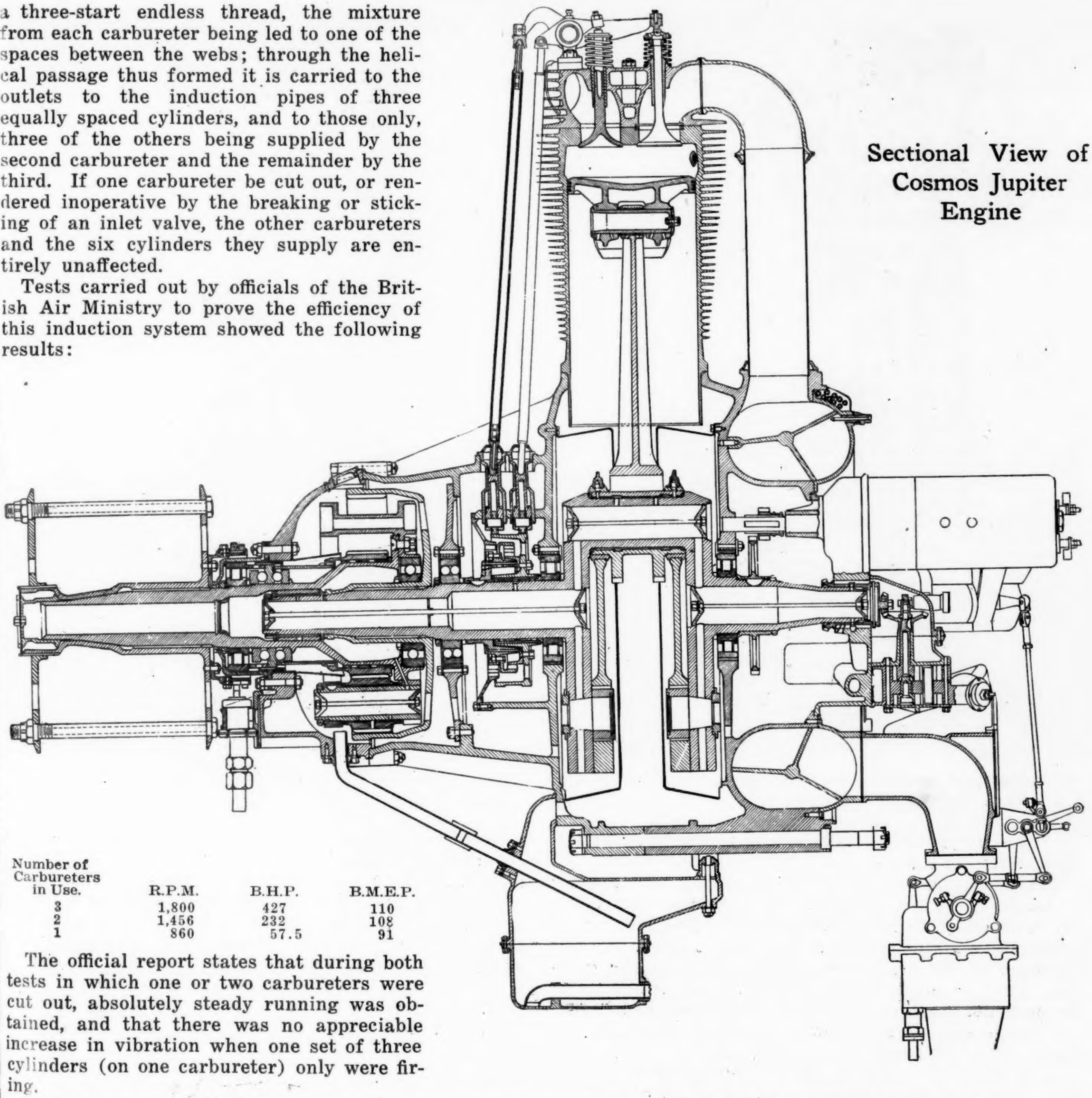
The crankshaft is bored as to both journals and pin and is supported by one ball and two roller bearings, the latter being against the crank cheeks. Oil is fed to the interior for conveyance to the big-end bearing by way of a supplementary white metal bushing at the rear end. For the pistons and piston pins, splash lubrication is depended upon, but the oil is led under pressure to the distribution gearing.

A small sump is secured below the crankcase, the oil draining into it through a 1-in. hole bored through the case at the front end, between two of the cylinders. To prevent oil thrown off by the big-end bearing of the master rod flooding the cylinders, each side wall of the crankcase carries two concentric troughs which collect the greater part of the surplus and lead it to the bottom of the case and to the sump; from the lowest point of the latter the scavenging pump draws the oil and delivers it to the main supply tank.

Three carbureters are used, supported by the elbows of an exhaust heated manifold of cast aluminum at the rear end of the crankcase. From the carbureters the mixture is led to a concentric chamber cast with the crankcase, and herein lies a special feature. Within the chamber is a circular cast aluminum distributing member, which takes the form of three webs radiating from a common center and arranged as a helix around the interior of the chamber. In effect it resembles

a three-start endless thread, the mixture from each carbureter being led to one of the spaces between the webs; through the helical passage thus formed it is carried to the outlets to the induction pipes of three equally spaced cylinders, and to those only, three of the others being supplied by the second carbureter and the remainder by the third. If one carbureter be cut out, or rendered inoperative by the breaking or sticking of an inlet valve, the other carbureters and the six cylinders they supply are entirely unaffected.

Tests carried out by officials of the British Air Ministry to prove the efficiency of this induction system showed the following results:



Sectional View of Cosmos Jupiter Engine

Number of Carbureters in Use.	R.P.M.	B.H.P.	B.M.E.P.
3	1,800	427	110
2	1,456	232	108
1	860	57.5	91

The official report states that during both tests in which one or two carbureters were cut out, absolutely steady running was obtained, and that there was no appreciable increase in vibration when one set of three cylinders (on one carbureter) only were firing.

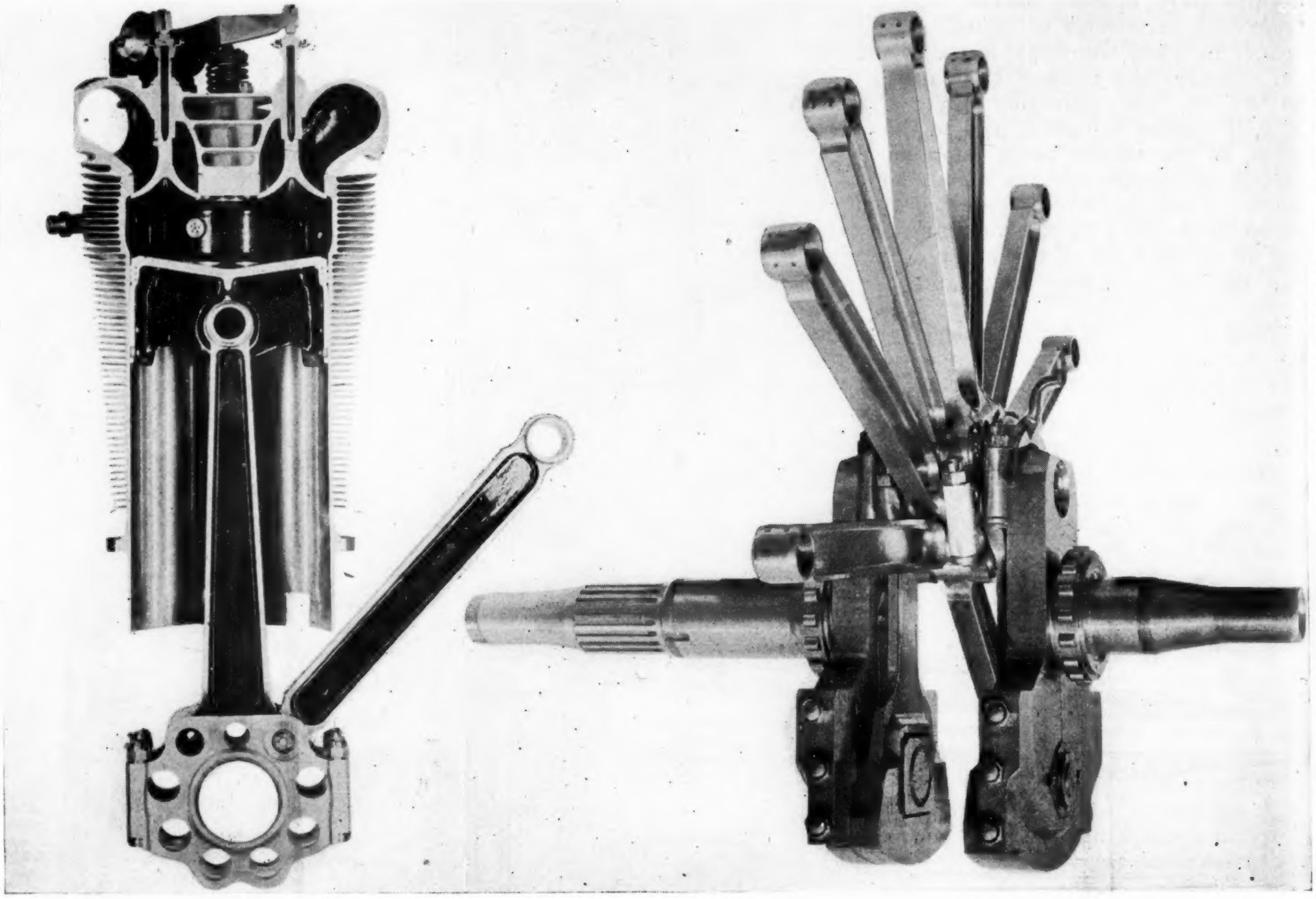
So far as carburetion is concerned, therefore, the engine may be looked upon as three three-cylinder units; any one or combination of which will run independently.

Another special feature of this engine which has already been alluded to, has been evolved to counteract the effect upon the under surface of the crank pin of attenuated cylinder charging and reduced gas pressures at high altitudes. The fact that, even under normal conditions of use, the under side of a crank pin is subject to greater wear than the top surface is generally recognized; this wear is, however, intensified when the cylinder charge is attenuated in a rarefied atmosphere owing to the inertia forces then having greater effect by reason of their being subject to less resistance from the reduced pressures above the piston.

To eliminate this cause of unequal wear under all conditions and especially when it is most pronounced, a pair of bob-weights are used to assist in holding the big-end bearing against the top of the crank pin.

One end of each bob-weight encircles and has a running bearing upon a side extension of the connecting rod big-end and cap; the other end is located alongside the crankshaft counterweight, of which incidentally, it serves as a portion. This outer end, as it may be termed, has a rectangular hole through it which fits over a square die or projection secured to the counterweight, the die having a longitudinal clearance within the hole. Thus, centrifugal force acting upon the heavy outer end of the bob-weight causes the inner end always to pull the big-end bearing in one direction, viz. against the top or outer surface of the crank pin and counteracting the inertia forces arising from connecting rod and piston.

While, as stated, this arrangement has been adopted primarily to counteract conditions which occur to a more pronounced extent at high altitudes, it appears to the writer to be a desirable fitting under all conditions of use; not only, too, for airplane engines, but in those of really high-grade cars—that is, if the advantage claimed



Photographic section of Cosmos Jupiter cylinder, piston and valves

Crankshaft and connecting-rod assembly of Cosmos Jupiter engine

for it should prove to be attained in practice and be evident after prolonged use.

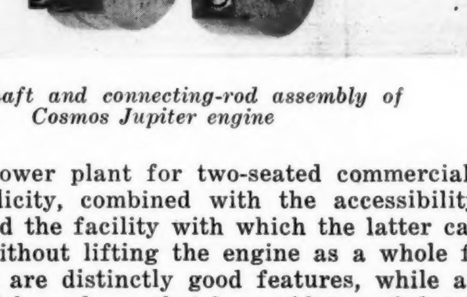
The engine described in the foregoing—the Jupiter model—is made in two types: (1) with direct drive and (2) with epicyclic reduction gearing. The latter type is identical in all respects except that it has a forward extension of the crankcase enclosing the epicyclic gearing. The prolonged driven member of the latter forms the propeller shaft and takes two bearings—one ball and one plain—on the crankshaft extension which in the direct drive type forms the propeller shaft.

Ignition is by magneto, two nine-cylinder machines, with pilot supports, projecting at an angle of 45 deg. from the rear extension of the crankcase and being driven by bevel gearing from the pinion on the end of the crankshaft that serves for the oil pump. The high tension wiring is, of course, in duplicate and leads to two sparking plugs in each cylinder. Provision is made for an electric starter, the motor projecting rearwardly between the two magnetos and driving the crankshaft through direct reduction gears.

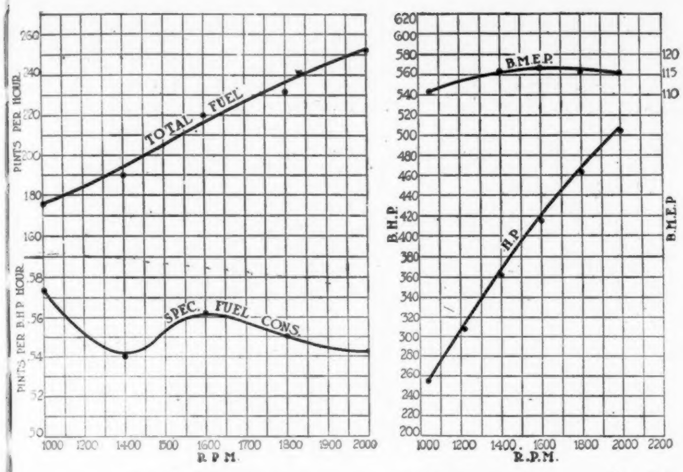
Reverting to the cylinder design, although the engines tested and approved of by the British Air Ministry have, as already mentioned, steel cylinders with integral and machined radiating fins, subsequent experiments have revealed that an improvement is secured with a finned aluminum jacket on a plain steel cylinder, in conjunction—as in the other arrangement—with an aluminum head.

The smallest model of the Cosmos series, the three-cylinder Lucifer, is a type that has proved itself admirably suited to training machines, and should be an excellent power plant for two-seated commercial planes. Its simplicity, combined with the accessibility of its units, and the facility with which the latter can be removed without lifting the engine as a whole from the machine, are distinctly good features, while as a production job, and one that has evidence of durability, it should appeal to students of airplane engine design.

Many of the details are interchangeable with those of the Jupiter, from which it varies mainly in having a shorter stroke ($6\frac{1}{4}$ in. as against $7\frac{1}{2}$ in.), no provision



The Cosmos helical distributor for the mixture from three carbureters



Performance curves of Jupiter engine

for bob-weights (obviously it is not a high altitude plant) and a countershaft train of pinions for the distribution gearing. Having only one carbureter the spiral induction system of the Jupiter is not required; in its place is a similar concentric chamber, into which the mixture is led, and in which is a baffle causing the gases always to flow in one direction before reaching the respective outlets to the three induction pipes.

The eighteen-cylinder, double row engine follows the Jupiter design very closely and has been evolved essentially for commercial purposes. It is designed to run at 1,750 r.p.m. with reduction gearing giving a propeller speed of 1,150 r.p.m. Duplicated Delco ignition is being used. The two-throw crankshaft has its pins at 180 deg., and balance weights solid with the shaft. The spiral induction system of the Jupiter is used. This engine has also the same type of valve gear, cylinder construction, lubrication system, and connecting rod assembly.

The following details of the Jupiter I (direct drive) engine will be of interest:

- No. of cylinders, 9.
- Bore, 5¾ in. (146.05 m.m.)
- Stroke, 7½ in. (190.5 m.m.)
- Bore-stroke ratio, 1.305 : 1.
- Compression ratio, 5 : 1.
- Area of 1 piston, 25.967 sq. in. (.1803 sq. ft.)
- Total piston area, 233.703 sq. in. (1.623 sq. ft.)
- Stroke vol. 1 cylinder, 194.7525 cu. in. (.1127 cu. ft.)
- Stroke vol. of engine, 1752.772 cu. in. (1.0143 cu. ft.)
- Maximum piston speed, 2312 ft. per min.
- Normal b.h.p., 400 at 1,650 r.p.m.
- Maximum b.h.p., 500 at 2,000 r.p.m.
- Maximum B.M.E.P., 117 at 1,600 r.p.m.
- Normal hp. per cu. ft. cyl. vol., 394.3.
- Normal hp. per sq. ft. piston area, 246.
- Maximum torque, 1277.5 lb.-ft.
- Total weight, 700 lb.
- Weight per normal b.h.p., 1.75 lb.
- Fuel consumption, 0.557 lb. per normal b.h.p. hour.
- Fuel consumption, 0.56 pints per normal b.h.p. hour.
- Oil consumption, 0.070 lb. per normal b.h.p. hour.
- Oil consumption, 28 lb. per hour.

Tests of Fokker Airplane Chassis

TESTS for strength have been made at McCook Field on the chassis of a Fokker airplane surrendered by Germany in accordance with the terms of the Armistice.

The chassis struts are seamless steel tubing of streamline section. Into the upper end of the strut is welded a ball which fits into a cup on the longeron. A bolt passing through both ball and socket holds the strut securely in place.

The struts are welded at the lower end to a box-shaped structure which also forms the support of the shock absorber. To this box is riveted an aluminum alloy cross member. The axle, which is 2 11/64 in. outside diameter, 0.160-in. wall steel tubing, moves inside this cross member.

The axle fairing has a section similar to that used for the wings, so that it acts as a lifting surface, contributing about 4 per cent of the effective lift.

Spiral steel springs, built up and wound on the tubes and axle in a manner similar to that employed with elastic cord, furnish the shock-absorbing medium.

The chassis, complete with wheels, tires and axle fairing, weighs 110 lb.

The action of the coil spring in absorbing shock does not seem to be as satisfactory as that of elastic cord. It was noted that when the load was removed the axle returned immediately to its original position, in contrast with the lag noted in tests where elastic cord was used.

The strength of this chassis is well above the average for airplanes of this type. The unusually high strength, moreover, is obtained with only a slight increase in weight. Allowing 10 lb. for the weight of the lifting surface construction, the net weight of 100 lb. is the same as that of the Ordnance Type "D" and only 10 lb. more than that of the Curtiss-Kirkham biplane.

Bearing Metal Alloys

THE cost of a bearing metal depends very largely upon its tin content, as tin is the most expensive of the major components. Therefore, while a high tin content makes for excellence in bearing qualities, there has been a decided tendency to reduce the tin content of the Babbitt formula or replace some of the tin by less expensive metals, notably lead. The scarcity of tin in various countries during the war stimulated research on bearing alloys and some notable results were accomplished. The original Babbitt formula is 89 per cent tin, 7.5 per cent antimony and 3.5 per cent copper. This tin content is exceeded in the International Aircraft Standards Board's specification, which calls for 91 per cent tin, and 4.5 per cent of antimony and copper each, while it has been reduced somewhat in the S. A. E. specification No. 24, which gives proportions as 84 of tin, 9 antimony and 7 copper.

One advantage of the true Babbitt and the Aircraft Board composition over the S. A. E. alloy is that they are more fluid and therefore can be cast in thinner shells.

The favorite Babbitt-Ersatz of the Germans during the war was made up of 21.3 tin, 3 copper, 63.3 zinc and 12 lead. These proportions were determined by scarcity of both tin and copper. Experiments by the Bureau of Standards have shown that results equal those obtained with genuine babbitt can be obtained from a composition of lead with a slight addition of alkali or alkali earth metal.

Factors of Design in the Construction of Gas Engines

The engine designer is often faced with problems that call upon his own experiences as a guide to the solution. This article was written by a man who has built engines for twenty years. He relates briefly some of the difficulties he has encountered and shows how they may be overcome.

By E. W. Roberts*

THE following more or less random notes are culled from the experience of the writer during a period of over twenty years devoted to internal-combustion engines and their design. Marked advances in internal-combustion practice have been made during this period, but the writer finds that the basic principles underlying internal-combustion engineering at the beginning of the present century are generally applicable to-day.

We have simply broadened their application, learned more about properties of materials and the performance of engines under special conditions. For example, we pride ourselves considerably on our developments in airplane engines. We have steel cylinders, light reciprocating parts and many other features which enable us to secure high power with a minimum of weight. Yet, twenty-six years ago Maxim built an airplane engine with steel cylinders having walls but 3/64 in. thick, with flanges much like those on the cylinders of our famous Liberty engine. Maxim's was a steam engine, it is true, but the principle of weight reduction was the same as we know to-day.

Reference to Books.—The designer, especially one who is called upon to design engines of a wide variety of sizes to meet a diversity of conditions, makes frequent reference to books and records of existing designs. He finds that the majority of published data consists of reproduced manufacturers' drawings and tables of sizes of parts, none too comprehensive in their character. If he seeks equations upon which to base computations of various parts, he finds the greater bulk of this material clouded with a mass of mathematical discussion, more confusing than otherwise. Even to the university man recently graduated, this mass of figures is appalling.

A great deal of the published material on design of a mathematical character is prepared by educators, frequently men of limited practical experience. Much of this matter is incomplete and I regret to say that quite a little of it is unreliable, and even misleading. I have in mind a book on design written some years ago, in which the valve settings were ten years behind the practice of the year in which the book was written. Books written from the standpoint of the university classroom are occasionally, but not usually, of value to the man who must design an engine for actual production in the shop. Books prepared by practical men are more likely to contain examples of existing designs and tables of sizes but no equations by which to compute the properties of engine parts. Occasionally, but not often, a designer will find in such books a drawing of a shaft or a connecting rod which will fit the engine he is laying out. Yet the best foundation for the design of any part is experience. Frequently we must depend for the desired information on the experiences of others. If we cannot find a part used on another engine which will meet the requirements of our engine design, what should we do? It is the answer to this very question that forms the basis of the following discussion.

Collecting Drawings.—To begin with, the engine designer should fortify himself with as much information as possible in the form of drawings of existing engines. These drawings can be secured in various ways, many of them from periodicals and recent books. He will find a wide variation in practice, no two engines of similar piston displacements having shafts of the same diameter, valve or port areas alike, etc. They cannot all be right.

Then what particular engine should be taken as a model? This is often a vexing question. The best clew to the solution of this problem is the selection of an engine manufactured by a concern long in business and which can reasonably be expected to have encountered the greatest amount of practical experience, especially to have taken advantage of difficulties encountered in use and to have overcome them. The designer must learn to weigh carefully the evidence in favor of certain features shown in various designs and to select those employed by the greater number of makers. This evidence, if possible, should be supplemented by a knowledge of the performances of the various engines in the field and of the particular features causing trouble.

From the information secured from existing designs, the designer will find he can determine factors for the design of any engine of the same class. In fact, he may be able to evolve master drawings from which, by simple proportion, he can quickly determine the sizes of the parts for any engine of the same class.

Crankshaft Design.—To illustrate, let us take the crankshaft. The basic equation for the determination of the diameter of this part is

$$s = k^3\sqrt[3]{m\,L\,D^2}$$

Wherein

s = diameter of shaft in inches;
k = a constant;
L = length of stroke in inches;
m = maximum pressure of the explosion;
D = diameter of cylinder in inches.

In this equation, we have the two factors L and D which are engine dimensions. The factor m is determined

Mr. Roberts is a consulting engineer and designer.

from the conditions of engine operation and k varies with the material in the shaft. We can combine m and k for engines operating under similar conditions and secure an equation of the form

$$s = K \sqrt{LD^2}$$

We can then call K a factor of experience and determine this from existing designs of engines of the same class.

In oil engines, particularly those of the "semi-Diesel" and similar types, we must take K much larger than in gas or gasoline engines. Shafts for engines of this type, which are based on gas engine practice, broke with appalling frequency in use. The cause was laid to the tendency of these engines to preignite and to build up a powerful back pressure during the compression stroke.

In certain recent designs of heavy duty automotive engines, particularly those for truck and tractor use, the factor K is quite large. In this case, the underlying reason is the desire to reduce the vibration-in-mass, as well as to reduce bearing pressures without increasing the bearing length. No amount of mathematical gymnastics, resulting in a hysteria of differentiation and integration, will give as satisfactory a value for K as experience.

From experience, we can check back to determine the factor of safety which was employed, especially if we know the grade of steel used. We can then evolve a satisfactory equation for determining the proportions of the crank arms. However, a simple proportion of the width and the thickness of the arms with the diameter of the shaft as a unit will usually answer every purpose.

Determination of Bearing Length.—The determination of the bearing length is a combination of two factors, one the pressure per square inch of projected area and the other the speed of the engine. At moderate speeds, it is sufficient to allow 400 lb. per sq. in. of bearing surface for ordinary methods of lubrication. At high speeds, the length of the bearing must be checked by the equation

$$l = \frac{APR}{X}$$

Wherein

l = length of crankpin in inches;
A = area of piston;
P = mean effective pressure;
R = r.p.m.;
X = a constant.

The value of X will vary with the method of lubrication and ranges from 400,000 for ordinary methods of splash lubrication to 500,000 for copious splash and 650,000 for pressure feed oiling. The value of X should be determined from existing designs of engines which operate under similar conditions to that under consideration.

One of the greatest faults in automotive engines has been short crankpins. This has been due to the fact that the length has been based upon allowable bearing pressure, ignoring the relation between bearing length and speed. Again in this case, the factor X becomes a factor of experience. If X is chosen too small, the bearing will heat. In this case, as in the computation of other parts of the engine, there are two methods of computation, one of which must be balanced against the other and the larger result employed.

Connecting Rods.—In the design of the connecting rod, we have a double computation necessary to determine the size of the rod. The usual computation is made to determine its resistance to bending, employing Euler's formula. Short rods of low strength material, such as malleable iron or low grades of cast steel, will fail if designed solely to resist bending. I know this from sad experience. The designer should determine the cross-sectional area of the rod and check its resistance to crushing by means of the equation

$$a = \frac{Am}{f}$$

Wherein

a = area of rod in sq. in.;
m = maximum pressure of explosion;
f = safe load.

The following table gives the value of f for various materials:

Material	f
Phosphor bronze	5,000
Manganese bronze	6,600
Cast steel	6,200
Low carbon steel (20 point)	6,000
High carbon steel (40 point)	8,800
Alloy steel (heat treated)	13,500

Naturally the safe load will vary with alloy steel, but generally the value determined by Euler's formula is higher for alloy steels than for resistance to crushing.

The practice occasionally followed of drilling holes in the web of a rod of I-beam form will not always stand up under an analysis on the basis of resistance to compression. Much has been written on balancing to reduce torsional vibration of the crankshaft. Stiffening the shaft will often overcome this effect in a much simpler manner than by a complicated system of balance weights.

Hollow Crankshafts.—If, as in the case of an airplane engine, increasing the diameter of the shaft makes it too heavy, the shaft may be bored and yet retain a large percentage of its torsional resistance. It is surprising to an engineer who has not attempted it what a material reduction in shaft weight can be accomplished by boring. On a 4½ x 5 in. four-cylinder engine, I used a shaft 2½ in. diameter, 40½ in. long with a bore of 2¼ in., except at the propeller hub. The actual weight of the shaft as it left the grinder was 17½ lb. Over one hundred of these engines were in use without a single broken shaft.

While on the subject of balancing, I may pause to present a method of computing balance weights which I have used with excellent results on engines running under 2000 r.p.m. The method is to balance all of the rotating weight plus half the reciprocating weight. In the case of the connecting rod about five-eighths of the rod may be considered as rotating. Hence the balance weight should exactly balance the shaft, if a weight equal to half the sum of the weights of the piston and the small end of the rod, plus all of the large end of the rod were to be hung on the crank pin. In other words, half the weight of the piston and five-eighths of the weight of the rod is considered as acting at the crankpin.

Balance weights should, if possible, be attached to the crank arms. If placed in the flywheel, as is done in many stationary engines, it is necessary to use an extra stiff shaft or the unbalanced flywheel will spring the shaft.

Three-Cylinder Engines.—Theoretically, a three-cylinder engine with crank pins at 120 deg. interval is inherently balanced. In a two-cycle, wherein the impulses come at regular intervals, this arrangement should prove ideal, but there is a certain amount of end sway. However, a six-cylinder, two-cycle with crank pins 1, 2, and 3 and 4, 5 and 6 at 120 deg., and crank pins 3 and 4 at 60 deg., gives ideal performance so far as balance is concerned. The most severe test, the operation of the engine in a light airplane, showed practically no vibration in flight. To use the expression of the aviator, "You could not see a wire shake."

In the matter of valves, their size and timing, we find a wide variation in practice on various engines. The

tendency, especially in recent designs of truck and tractor engines, is to use large valves with small lifts. The old idea was to use a speed in the inlet passages and through the valve of not to exceed 100 ft. per second. To-day, we find velocities of fully three times that speed.

Inlet Gas Speeds.—In a recent discussion of inlet speeds, great stress was placed upon the necessity of producing turbulence in the mass of the mixture in the combustion chamber. The size of the valve is based on the desire of the designer to get the greatest possible volume of air into the cylinder,—in other words, to secure the highest possible volumetric efficiency. Not all designers realize the importance of this and wonder why their engines are deficient in power output, especially at high speeds. It may surprise the reader to know that volumetric efficiencies vary from 50 to 85 per cent.

In reference to valve openings, the average rule-of-thumb designer assumes the valve to be open at its full lift during the entire suction stroke. As a matter of fact, the "dwell" or the period during which the valve is fully open is less than 20 per cent of the time elapsing while it is off its seat. The balance of this period finds the valve opening restricted in area. Probably as good an example as possible of modern practice in valve proportions is to be found in the Liberty engine. In this engine the bore is 5 in., the stroke 7 in. and the inlet valve is 2½ in. in diameter, with a lift of 7/16 in. The area past the valve at maximum lift is, therefore, 4.82 sq. in. Based on the customary hypothesis that the volume of the air entering the cylinder is the product of the area of the piston by the mean piston speed, we have, at 1700 r.p.m., a speed through the valve opening of 134.5 ft. per sec.

In certain modern designs of tractor engine, we find in a 4¼ in. by 6 in. engine, 2 in. valves with ⅜ in. lift or practically 100 ft. per sec. speed through the valve at 1000 r.p.m. This would show that American engine designers, at least, are still holding close to the old established rule of 100 ft. per sec., which is actually a hold-over from the steam engine.

Design of Valve Springs.—In the design of springs for valve return, designers have, quite frequently, overlooked the actual factors bearing upon the subject. To begin with, they have based their computations on the average speed of the valve in closing. This is a case where rule-of-thumb methods are inadequate. The spring must make the valve follow the cam where the speed is greatest. In order to determine this speed, it is necessary to lay out the valve velocity diagram, and from this diagram secure the maximum speed of the valve. Again, the inertia of the valve alone is not sufficient for consideration. The weight of the push rod must also be made a factor in the computation. At once, the designer will see the drawback to the use of a heavy push rod or other valve operating mechanism.

Neglect of the valve operating mechanism in the computation of the spring accounts for much of the valve noise which develops at high speed. Another drawback to heavy valve mechanism for high speed engines is the heavy pressure on the cam, resultant from the use of adequate springs.

Mushroom Type Push Rods.—While on the subject of valves, I might mention, in passing, that I have used the mushroom type of push rod on all of my designs for the past fourteen years, wherever the lubrication was ample. It is the custom of some designers to place the center of the cam to one side of the axis of the push rod extended. The idea is to make the cam turn the push rod on its axis. I defy anyone to so line up a mushroom type of push rod and its cam that it *will not* turn. The slightest possible rotation is sufficient to distribute the wear and this is secured with the most careful alignment, so why offset them? The advantages of this type of push rod over the roller type are, simplicity of manufacture, lack of roller troubles, no need for a guide to prevent rotation, and a much better action on the valve. If properly hardened, the wear is inappreciable.

Valves with narrow seats, scarcely 1/16 in. for a 1½ in. valve, have been used with success by some engine builders. The old 45 deg. seat is almost a fixed idea among engine builders, but the 30 deg. seat has shown excellent results, as, for example, in the Liberty engine.

Foundry Limitations.—Absolute ignorance is displayed by certain designers in regard to foundry limitations. Cores are made of wafer thickness and of metal as low as 1/16 in. in iron castings and pockets so located that it would be impossible to vent them. My experience is that the limit of the iron founder is ⅜ in. cores and 5/32 in. walls. Furthermore, it takes a good foundryman to go to such limits.

In designing a cylinder or a cylinder head, be sure of two things: Take the water out from the highest part of the jacket to avoid a steam pocket and provide ample water around valves and spark plugs. See that your water passages are so arranged that in passing from the inlet to the outlet, the water must flow through all water spaces. Watch out for dead ends or "blind alleys." I saw a violation of this rule that nearly put an automobile company on the rocks. It was a four-cyclinder en-bloc casting in which the water entered at the base of the front cylinder and was taken from the top of the same cylinder. In consequence, the water followed the shortest path, which was through the jacket of the front cylinder, and the rear cylinders ran hot.

It took automobile engine designers a long time to learn that they must avoid making the walls of adjacent cylinders touching or integral. The trouble with this construction is that the cylinders will go out of round and the gas will go by the piston. Such an engine will fall off in power as soon as it warms up. In a two-cycle engine, the water jacket must be carried around the cylinder beneath the exhaust port.

Two-Cycle Engines.—The two-cycle is making a slow recovery in the automotive industry. Note the inroads of this type in the motor-cycle field. With one exception, all "semi-Diesel" engines are of the two-cycle type, and two-cycle Diesels are quite plentiful.

About half of the engines I have designed have been two-cycles. They are a mystery to some, a *bête noir* to others. However, the salient features of two-cycle design can be covered in one short paragraph. In a three-port two-cycle, open the exhaust at 60 deg. in advance of the dead center, the inlet from the by-pass at 45 deg., and the inlet to the base at 44 deg. for a four-cylinder or 37 deg. for sweet running when throttled down. The areas of the ports should be in proportion to the time the port is open. The exhaust port, for example, is open 120 deg. The area should be at least 1½ times the area of the exhaust on a four-cycle engine. The inlet to the cylinder should be the same angular length as the exhaust. This gives a small area but the pressure in the base makes up for it. It usually requires at least twice as big a carbureter (in area) for a two-cycle as for a four-cycle for the same piston displacement and the same speed. Make the crankcase hug the crank arms and the rod, giving not over ¼ in. clearance, to secure ample base compression. Lubricate by mixing the oil with the gasoline, one part of oil to sixteen of gas. There is no detriment in using base compression, popular opinion to the contrary notwithstanding. Flexibility in a two-cycle may be made fully equal to that of a four-cycle by proper port design.

Uniform Dimensions Would Reduce Truck Building Costs

Standardization of truck practice is a vital need in the industry to-day and Mr. Schipper is pointing out the way in a conclusive manner. This article relates to the mounting of the body or cab and to arrangements for the power take-off, suggesting a few possible uniform dimensions.

By J. Edward Schipper

IMMENSE sums could be saved truck users if greater uniformity prevailed in a number of the chassis dimensions. The points particularly in mind are the control units and the dimensions having to do with the mounting of the body or cab and with the arrangements for the power take-off. The motor truck industry is saddled with a tremendous expense, due to the fact that bodies and other necessary units must be made up specially after the chassis itself has been sold.

The result is that manufacturers cannot start work until they receive a definite order and in consequence are unable to obtain continuity and uniformity of production in their plants. In other words, they do a job business in place of a steady productive business, which could be run to greater profit and advantage, and at the same time would reduce prices to the consumer.

It is admittedly a fact that, if truck prices were lower, more trucks would be sold and there is no better way in which truck manufacturers could go about bringing down prices than by getting together on a number of points having to do with cab and body dimensions. Were it possible to bring about a set of recommended dimensions which could be used exactly or approximately by a large number of truck chassis makers, it would be possible for the body, or the cab manufacturers, to produce on a better economic basis and, consequently, to quote lower prices.

A canvass of the truck body and truck cab manufacturers indicates that a movement toward something like standardized or recommended dimensions for truck chassis parts which influence the dimensions of the cab or body would be hailed with enthusiasm. This is not a new thought, as body manufacturers have from time to time taken it up with the truck builders, and a limited degree of standardization prevails. There is not, however, a generally accepted set of recommended dimensions for parts which influence the product of the body manufacturers and, consequently, matters are quite chaotic when it comes to the purchase of special cabs or body equipment. In certain sections of the country, where it is necessary to order from body and cab builders long distances away, it becomes an expensive proceeding.

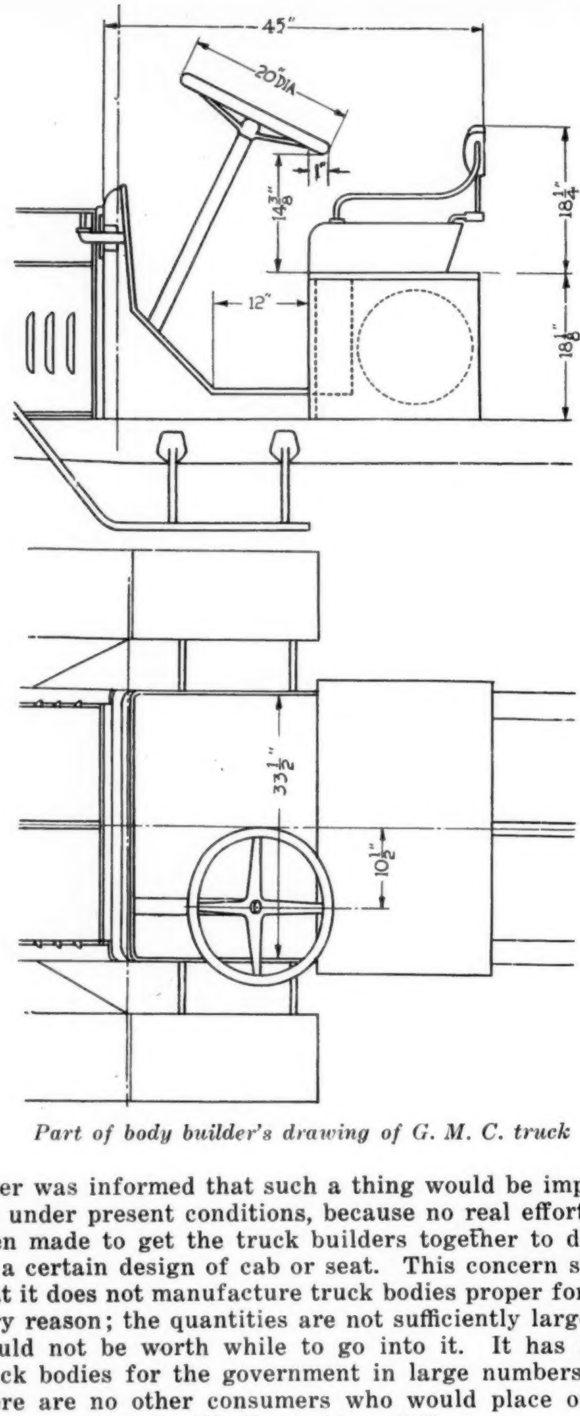
A study of the dimensions and locations of control units on a large number of trucks indicates that in many instances dimensions vary by only a fraction of an inch, and, in fact, the differences between the extreme dimensions determining the location of the front seat, steering column, brake and shifter levers and the pedals, is not as wide as would be supposed. This is evidenced by the tabulation herewith.

Truck manufacturers are not as enthusiastic about standardization as are the body builders, because they are not directly affected. Nevertheless, if a movement did result in establishing standards on certain parts and recommended practices on others so that it would be possible for the body and cab makers to approach something like a production basis, it would act to the benefit of the truck manufacturers, who would be able to increase their sales as a result of the lower prices.

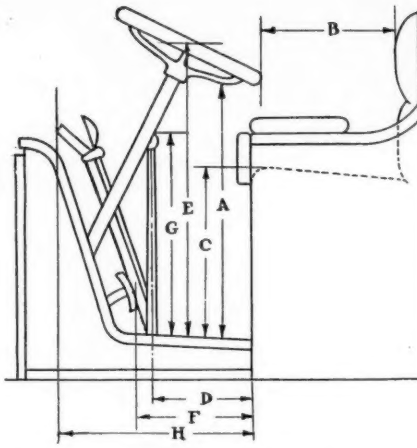
At first sight, there are some engineering reasons why it would not seem practicable to attempt to recommend a set of dimensions for a given capacity truck. As one manufacturer expresses it, "It will be utterly impossible even to think of standardizing the positions of the brake, clutch and accelerator pedals. These locations are never determined from the ideal standpoint; the pedals are put in wherever we can get them in, as they have to clear other units under the floor boards. The pedals are usually left until the last in laying out a new job and the result is that they are usually pretty crooked affairs. There is no one point which gives us so much trouble as this location. There is no difficulty in locating the steering wheel in the desired place, and our rule is that the back edge of the steering wheel should be 14 in. above the top of the board on which the seat cushion rests and 1 in. back of the front edge of this board. These dimensions are worked to, irrespective of the angle of the steering column, and seem to make for a satisfactory location."

Even this manufacturer, who sees considerable difficulty in the way of standardizing control unit locations, adheres to one standard as regards location of the steering wheel. If only this single dimension were standardized, it would permit of a standard seat height for a given capacity, and would relieve the body or cab manufacturer of worry on this score.

As a matter of fact, there are sheet steel workers who would like to get into the body-making business, but who have been kept out of it on account of the necessity for making every job special. One Philadelphia concern, which is in the sheet metal business, states that it has received inquiries from truck manufacturers for dash seats or cabs to be made entirely of steel, but individual orders are not big enough to warrant the making of dies for these parts. Each truck manufacturer, it seems, has a different idea regarding the dimensions which would affect the steel seat. This concern recently had an inquiry from a fairly large truck plant asking if a standard seat could be built to fit all the trucks. The manufac-



Part of body builder's drawing of G. M. C. truck



Sketch showing the seat and control dimensions and location of the Federal truck

DIMENSIONS IN INCHES

MODEL	SD	TE	JE	WD	XD
Tons	1	1½	2	3½	5
A	28	26½		29	
B	16½	15		16	
C	19	19		20	
D	11	10		9	
E	32	31½		31	
F	14	15		14	
G	22½	22½		22	
H	25	23		23	

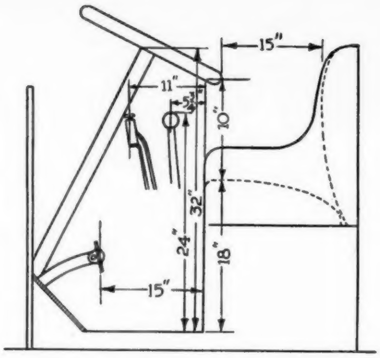
turer was informed that such a thing would be impossible under present conditions, because no real effort had been made to get the truck builders together to decide on a certain design of cab or seat. This concern stated that it does not manufacture truck bodies proper for this very reason; the quantities are not sufficiently large. It would not be worth while to go into it. It has made truck bodies for the government in large numbers, but there are no other consumers who would place orders large enough to justify the making of dies.

With the increased use of all-metal bodies, the question of die making and die and tool cost for manufacture enters largely into the proposition and makes standardization of the chassis dimensions, which affect the body maker, of prime importance. Some of the points affected are the width of the chassis frame, the location of the control units, location of the gasoline tank, location of the battery, location of the tool box, location of the steering wheel, length of driver's seat or cab from dashboard to back of seat, measured from outside to outside.

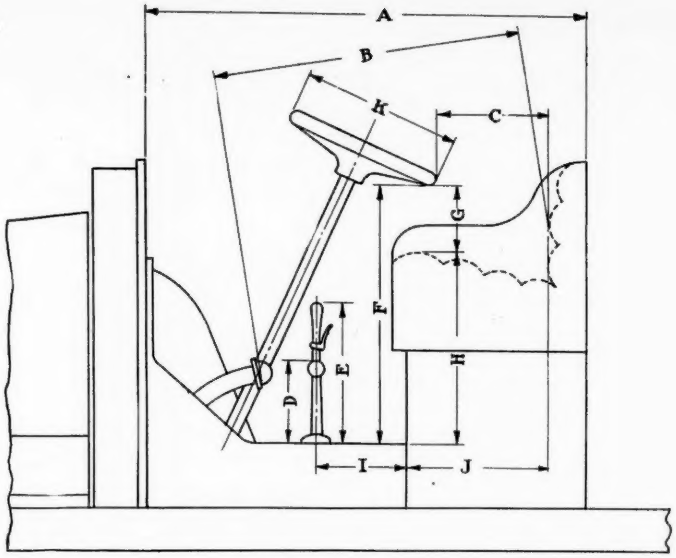
From the body makers' standpoint, something like uniformity in the control units would be a great help, because it would enable them to standardize the height of the seat, which has to be varied to suit the steering post. It would also make it possible to use a universal top, the peak of which had to cover the windshield and give a good, general proportion. Also, where the body manufacturer is asked to furnish toe sills and floor boards, this standardization would help to a large degree.

Dimensions in Inches Indicating

	Lewis-Hall Motor Truck Company		Gramm-Bernstein		Service Motor Truck Company
	2½-ton	3½-5-7-ton	Model 15-65-20	Model 25-35-50	
1—Steering wheel, lower edge to floorboard, vertically	34½	34½	29	26½	27¾
2—Steering wheel, lower edge to seat back, horizontally	18	19	19	19¼	$20\frac{5}{16}$
3—Height of front seat above floorboard, vertically	16	17½	19	17	18½
4—Shifter lever handle to front edge of seat, horizontally	11	7	8½	6	$8\frac{11}{16}$
5—Center of wheel to floorboard, vertically	31½	34	30	27¾	32
6—Clutch pedal to front edge of seat, horizontally	16	14	11½	14	15¾ approx.
7—Shifter lever handle height above floorboard, vertically	19½	21½	25	21	30½
8—Brake lever handle to seat, horizontally	11½ set	12 set	9½	17	13½ approx.
9—Transverse distance of sill to shifter lever	19	19½	14	15½	30
10—Transverse distance of sill to brake lever	20¼	20¾	12	11	36¾



Sketch showing seat and control unit locations of the Denby truck



Sketch showing seat and control unit locations on Kelly-Springfield truck

Capacity	A	B	C	D	E	F	G	H	I	J	K
1½ and 2½ tons.......	42⅝"	30"	13"	17¾	20¼"	27"	10"	17"	11½"	18"	20"
3½, 4, 5 and 6 tons....	45 7/16"	31"	13"	19½"	27½"	28"	11½"	16½"	12½"	18"	22"

The ideal would be the universal cab but this seems remote. There is no good reason, howevery, why it should be impossible, and the equipment manufacturers and jobbers would certainly be behind any movement towards fixing a set of recommended dimensions, if actual standards could not be adopted. Taking the single point of the dash alone, this is of the greatest importance to cab builders, for a dash of standard size would enable the production of cabs in quantities, instead of in twos and threes, as now produced. The location, position and size of the gasoline tank also play an important part in the standardization of cab and seat units, as do also the tool box, battery box and other attached parts in a great many instances.

The manufacturers of hoisting equipment are particularly interested in the adoption of a standard take-off to operate power hoists, winches and other loading and unloading devices. This, of course, comes back to the transmission gearset manufacturer and is in line with the movement which has already been noted, to establish a standard power take-off. One large manufacturer of dumping equipment writes as follows:

"The lack of standardization in this respect means that the ultimate user must pay considerably more for his dumping equipment or any special equipment that he has put on the truck. There is no reason whatever why the transmission people should not build a standard transmission from which extends a shaft to which any manufacturer of loading or unloading devices could attach his mechanism.

"The propeller shaft clutch, as it is commonly known, is unquestionably a detriment to the truck under ordinary conditions. The proper place to drive any device of this kind would be from a shaft extending direct from the transmission, having an idler gear in mesh with the proper transmission gear. While we understand there is a general move in this direction, it seems to us that this matter should be agitated strongly in order to accelerate its accomplishment. Every manufacturer of dumping equipment is intensely interested in this standardization. In the end the user pays for a lot of specialization and it throws that much more unnecessary burden on truck transportation."

The same situation confronts builders of other special truck units, whose parts must be made to conform to what the truck manufacturer has provided. There would be no better place to start than in the location of the controls. This is feasible because every truck is designed to be driven by a man of average physical size, and, consequently, practically all of the control units fall within a fairly narrow range of dimensions. The tabulations given herewith show this clearly and, where there are variations of from 1 to 2 in. in practically every case it would be possible for the manufacturers to come together and settle upon a definite location. The benefit to body builders, cab manufacturers, makers of hoisting and dumping apparatus would be incalculable, and, as has been strongly pointed out, the price to consumer would be less and would result in increased business.

(*Continued on page* 1347)

Locations of Typical Truck Controls

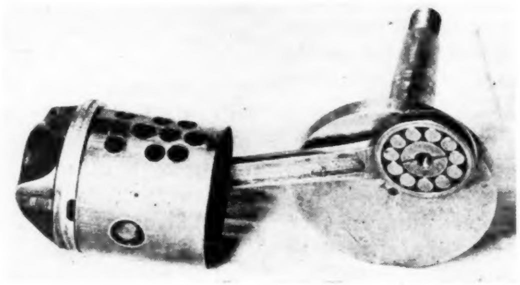
Republic Motor Truck Co.	Kelly-Springfield		Indiana Truck Corporation. Model					Clydesdale Motor Truck Company			Recommended
	1½ and 2½ ton	3½, 5 and 6 ton	12	20	25	35	50				
26¾	27	28	25	25½	27½	27½	27½	27½	29½	27	28½
15½	13	13	24½	23	22¼	23½	21½	17½	18½	15	17
20½	17	16½	21	21	21	21	21	20	20	20	20
8	11½	12½	10	7	7½	7	9	10	5	6¾	8
34	..	..	33¾	32	33½	33¾	32	33	33½	32	33
14½	12	13	14¾	16	16	15½	13½	15	13¼	13	14½
22½	17¾	19½	21¾	20	20½	20	20¼	26	21	20½	22
13¾	11½	12½	19½	19½	19½	18	21	18	18	..	18
18½	..	..	15¾	20½	20½	20¾	20	..	..	..	..
20¾	..	..	13½	22¼	22¼	22½	21¾	..	..	..	..

New Lightweight Motorcycle Shows Design Development

This article describes a new type of motorcycle developed in England having many novel features of construction and control. The design is based on ease of construction and maintenance and employs pressed steel shapes for the frame and parts enclosures. It is unlike American practice.

By M. W. Bourdon

ENGINEERING details of the Pullin lightweight motorcycle recently put forward in England represent a laudable attempt to depart from the stereotyped principles of construction. If the Pullin has not exactly the workmanlike appearance which is so much desired in motorcycles, it certainly exhibits a trend in the right direction. As will be seen in the illustrations, there is a comparative cleanliness of outline and an absence of the usual excrescences represented by exposed units such as engine, gearset, magneto and carbureter; notwithstanding this enclosure of main units, accessibility has not been sacrificed to any degree.

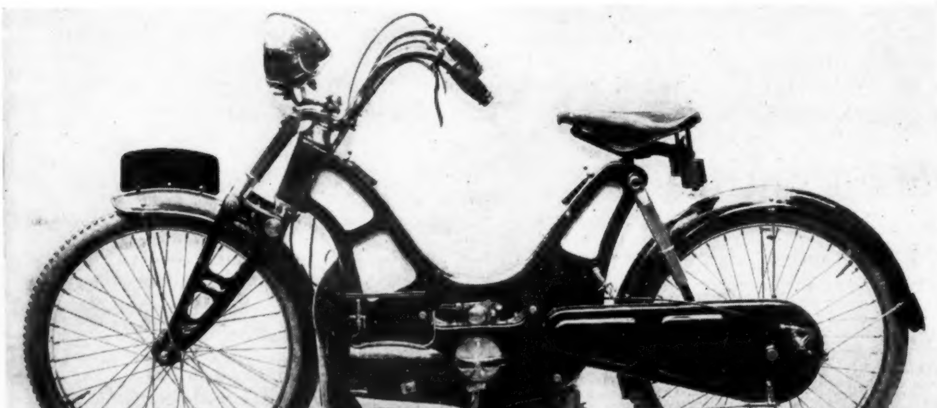


Pullin piston, connecting rod, roller big-end bearing and crankshaft

The usual tubular members forming the frame have been almost entirely eliminated in the Pullin. The main frame consists of two sheet steel stampings welded together at the longitudinal center line of the machine; the two "horns," which at their extremities support the saddle post and steering head respectively, form integral fuel and oil tanks, while at the bottom the two-cycle single-cylinder engine and the two-speed epicyclic gearset are secured in such a manner as to brace the frame units at this point and yet be readily detachable. There is a clear run for air draught between the frame members past the engine, which is horizontally arranged with horizontal radiating fins and has a detachable head attached to the unit cylinder and crankcase. Sheet steel air scoops integral with leg guards and foot-rests are fitted at the front.

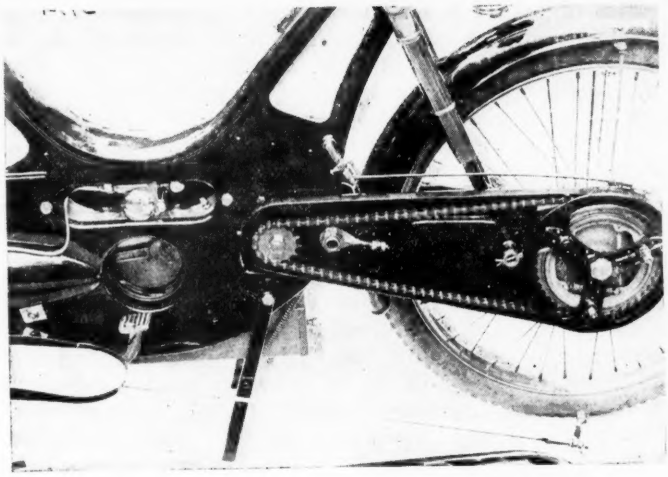
The triangulated forks are also of stamped steel, pivoted to the steering column with movement about the pivot center limited by duplicated helical springs enclosed within telescopic tubes; the latter in turn are pivoted, the top one to a bracket clipped to the upper end of the steering column. Thus sprung forks are provided, and a similar system of flexible mounting is used for the rear wheel forks and stays, one of the latter being of stamped steel and forming the inner member of the chain case for the final drive.

Besides this unusual frame construction, there are several engineering features of distinct novelty in motorcycle construction. For example, the engine, although it is of the two-cycle type, with crankcase compression and inlet and outlet ports in the cylinder wall uncovered by the piston at the bottom of its stroke, embodies a system of construction denoting an attempt to overcome the disabilities of the usual two-cycle motor in respect of slow running.



Left side of Pullin motorcycle; right side is even "cleaner," openings for crankcase, inspection plate and fuel mixer being absent

No carbureter of the accepted type is used, but in its place there is an air and fuel mixing valve held to its seat by a light spring. The fuel emerges to mix with the ingoing air through a small hole in the seat of the valve, this hole being normally closed by the mushroom head of the valve when the engine is at rest. There is no throttle in the ordinary sense, but a finger-operated needle valve permits of variation of the fuel supply to suit atmospheric conditions. A full charge is therefore drawn into the crankcase on each outward stroke of the piston and is



Rear part of Pullin frame with outer covers of chain case and crank chamber removed, disclosing connecting rod big-end, chain adjustment and expanding rear wheel brakes

there compressed, subsequently—at the end of the return stroke—finding exit through holes drilled in the piston wall. From these holes it passes into a transfer passage and into a cylinder by way of the inlet ports.

So far, there is no suggestion of throttle control. Constituting the latter is a valve in the detachable cylinder head; this is held positively to its seat when conditions represent "full throttle," but is opened more or less by the action of a helical spring when the handle bar control is operated to permit this movement. To reduce the charge, the spring is allowed to operate and open the valve; by the outward movement of the piston a certain proportion of the charge within the combustion chamber is then forced out past the valve and through an exterior passage back into the crankcase. Thus the amount of mixture in the cylinder at the time of ignition is varied according to requirements, and unless the engine is running at what corresponds to full throttle, the valve is constantly passing back to the crankcase a certain proportion of the mixture, the amount which thus escapes depending upon the extent to which the spring is released by the handle bar control.

This system, it is claimed, by maintaining at all times the full compression in the crankcase, eliminates one of the commonest faults of two-cycle engines, namely, "starvation" and low compression in the crankcase at small throttle openings, and the consequent inefficient transfer of the mixture through the ports into the cylinder.

The lubrication system is also entirely different from that of the usual two-cycle engine. The stem of the fuel mixing valve is enlarged in diameter where it passes through its guide, and this enlarged diameter, in conjunction with the guide, forms an automatic piston valve which allows lubricating oil by gravity to run through a small orifice when the mixing valve is lifted from its seat. The small amount of oil thus allowed to pass is carried to the upper side of the cylinder wall, lubricating the latter and the piston, the surplus finding its way into the crankcase and draining into a small sump. The oil which accumulates in the latter is circulated by the crankcase compression to the single journal bearing of the crankshaft and thence to the epicyclic gearset, lubricant that exudes from the end of the bearing overflowing on to the primary chain sprocket and into chain transmission case.

The lubricating system is thus entirely automatic, with the exception of a two-way tap opening or closing the exit from the tank and a maximum feed regulating valve.

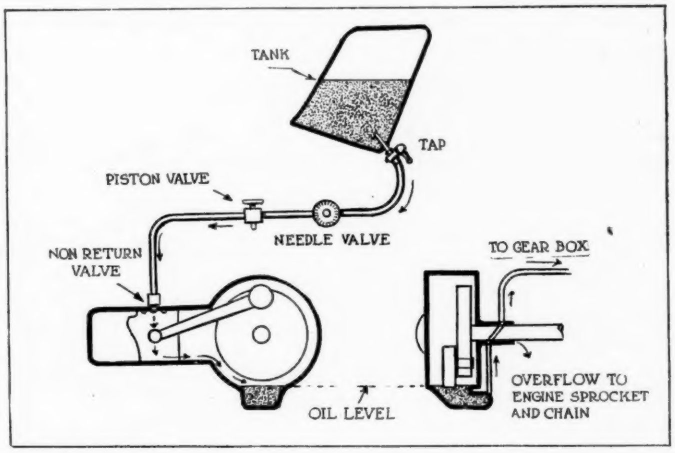
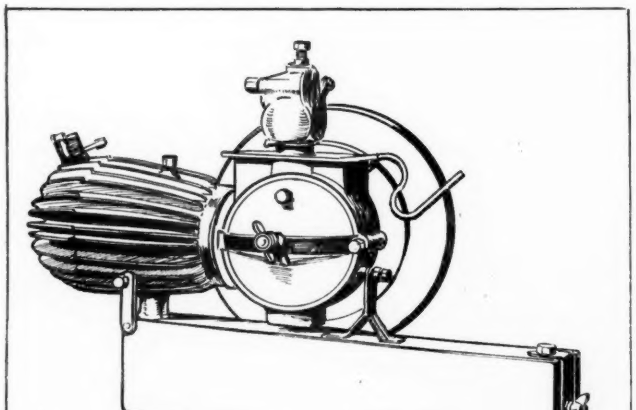


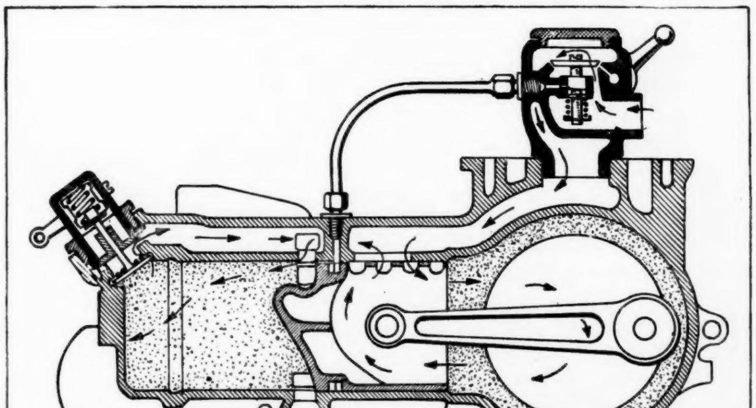
Diagram of the Pullin motorcycle lubrication system

No direct lubrication is provided for the big-end bearing, which is of the roller type, the pin being formed integral with the disk which serves as the crank web.

A separate magneto has also been dispensed with, ignition being provided by a flywheel magneto having a con-



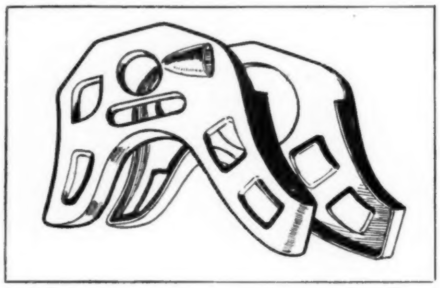
Pullin engine, fuel mixer and muffler. Cylinder is of cast iron as unit with crankcase; head is detachable



Diagrammatic section showing Pullin fuel mixing valve, cylinder discharge valve (displacing throttle) and circuit of mixture in crankcase, cylinder and transfer passages

tact breaker which can be made immediately accessible by the removal of an inspection cover. No handle bar control is provided for ignition timing, and although the latter can be varied by direct adjustment, it must, from the rider's point of view, be considered as fixed.

The spark plug is arranged in the cylinder head and projects at the front through the opening between the frame members. Through this opening can also be removed the detachable cylinder head, without disturbing the engine as a whole. If the piston also is to be detached, the big-end bearing can be cast adrift and the piston withdrawn through the front of the cylinder, the interior of the crankcase being rendered accessible by the removal of a quickly detachable side cover plate.



Stamped steel units forming Pullin frame; "horns" are utilized for fuel and oil tanks

The drive is taken from the crankshaft sprocket through an enclosed roller chain to the two-speed epicyclic countershaft gear. High and low speeds, respectively, have ratios of 6 to 1 and 12 to 1, the epicyclic gearing being of the bevel type, while the direct drive clutch and the low speed brake are combined as a sliding cone unit. From the gearset the drive is taken through another roller chain to the rear wheel sprocket, which is integral with the drum of the internally expanding brakes. There are three shoes for the latter and two operating spindles, two of the shoes being expanded by pedal control and the other by Bowden wiring from the handle bar.

Various detail refinements which will naturally suggest themselves have been embodied; for instance, accessible brake adjustments are in evidence, and only two standard sizes of nuts are used throughout. Then, too, front and back wheels are readily detachable and also interchangeable. With 24 x 2-in. tires, pressed steel carrier and metal tool-box (neither of the two latter is shown in the illustrations) the complete machine weighs approximately 130 lb., with a $2\frac{1}{2}$ x $2\frac{1}{2}$-in. engine developing 3 hp. at 1900 r.p.m. This represents an unusually low weight, for the average British two-cycle machine corresponding to the Pullin weighs from 170-200 lb. The fuel consumption is said to be one Imperial gallon to 110 miles, while the announced price without lamps is $250 at the normal rate of exchange.

The designer of this machine is a man well known in the British motorcycle industry, having been connected with some of the largest plants and possessing a reputation as a successful rider of racing motorcycles. Clearly, then, it is not the production of a designer without experience as to requirements and pitfalls.

Development of Electric Steel Furnaces

IN his presidential address to the Iron and Steel Institute (Great Britain) Dr. J. E. Stead referred to the rapid development of the electric steel furnace during the war. He said that this furnace, which, up to the outbreak of war, only produced a very small proportion even of the higher grades of steel, had developed with great rapidity during the last five years. Before the war progress had been most rapid in Germany, where conditions were peculiarly suitable to its use. Most of the German electric steel was made by refining basic Bessemer steel, and furnaces up to 30 tons capacity had been installed for this purpose.

The United States came second, and Italy, America and France were ahead of England in production. During the war great progress was made in England, and the production was now only surpassed by the United States and Germany. The latter still held second place, but only because most of the steel there was refined basic Bessemer steel, while British furnaces were in practically all cases melting cold scrap. The actual number of furnaces and the power used were greater than in Germany.

The product in England before the war was probably about 10,000 tons a year, though no exact figures were available. This was made in eleven furnaces with a total transformer capacity of about 4500 kilowatts. By the end of 1918 the number of furnaces of all types had increased to about 140, with a production of about 150,000 tons a year and a transformer capacity of nearly 100,000 kilovolt-amperes. The possible production at the present day, if all furnaces were worked to their full capacity, is about 300,000 tons a year. In 1918 a number of furnaces had either just started work or were not completed, and several were shut down immediately after the armistice was signed.

Stainless steel was being made in increasing quantities, and the demand for nickel chrome and other alloy steels for motor cars and airplanes was increasing again. On the Tyne electric steel was being made at a price which could compete with the acid open hearth, cheap power being available from coke-oven gas, and its entry into the market made it probable that the tonnage would soon be largely increased.

In America and Germany electrically refined Bessemer steel was largely produced, and it was possible that a duplex process would be the next large development here. An electric furnace refining basic open-hearth steel made from Cleveland ore could produce steel of the quality at present made in acid open-hearth furnaces at a cost which would show a substantial profit at present prices.

One of the latest advances in connection with electric steel was to melt it in an induction furnace *in vacuo*. Albert Hiorth, of Christiania, had designed a furnace particularly adapted to effect this object. The steel, after being melted *in vacuo* in the ring furnace, was allowed to cool down without admission of air into the furnace. The steel was then removed and cut up into sections for forging. By this means steel free from honeycomb and gases was obtained. It was, however, probable that the process would be applied only for very high qualities of steel.

THE list dated April 1, of Research Associations which have been approved by the Department of Scientific and Industrial Research (Great Britain) as complying with the conditions laid down in the Government scheme for the encouragement of industrial research and have received licenses from the Board of Trade include: The British Iron Manufacturers' Research Association, the Research Association of British Motor and Allied Manufacturers, the British Portland Cement Research Association, the British Scientific Instrument Research Association, the Research Association of British Rubber and Tire Manufacturers, the Glass Research Association, and the British Non-Ferrous Metals Research Association.

Acceleration of the Automobile and Its Measurement

This phase of automobile design and engineering, considering the present day conditions of high speed and multiple cylinder engines and increasing traffic congestion in the cities, has attained a high importance. Consequently, Mr. Heldt discusses it fully, showing particularly how to plot the acceleration curve of cars not yet built, giving the necessary formulae.

By P. M. Heldt

IN recent years considerable attention has been paid to accelerating ability in the design of passenger cars. On the one hand, the advent of multi-cylindered, high speed engines made high rates of acceleration possible and, on the other hand, increasing congestion in the main thoroughfares of our big cities, necessitating frequent stops and restarts, made it very desirable. Salesmen began to extol the accelerating powers of their cars by describing them as having "lots of pep" and in the course of time there arose a need for some means of measuring the acceleration.

Similar problems had arisen earlier in the railroad and street car fields and instruments known as accelerometers had been developed for the purpose, which show the acceleration directly. As is well known, acceleration is the rate of speed increase and as speeds of the order here dealt with can be conveniently expressed in feet per second, acceleration is expressed in feet per second per second. Therefore, the indications of an accelerometer are in terms of this unit. The force of gravity imparts to a body an acceleration of 32.16 ft. per second per second, and the unit of acceleration, one foot per second per second, is, therefore, only about one-thirty second that due to gravity.

Most of the instruments built for railway work are not applicable to automobile tests, mainly on account of their cumbersomeness and their sensitiveness to acceleration in other directions than the direction of vehicular motion. An automobile body while passing over the road is constantly swaying more or less from side to side, due to unevenness of the road surface and hence is subject to transverse acceleration; it is also being thrown upward by road obstructions, and, therefore, also accelerates and decelerates in a vertical direction. But it is only the acceleration in the direction of travel that it is desired to measure.

Wimperis Accelerometer.—In 1909 an instrument specially designed for automobile tests, known as the Wimperis accelerometer, was brought out in England, and has been used to some extent ever since. The original object of the designer seems to have been more to evolve an instrument suitable for determining the traction resistance of roads than one giving the accelerating power of a vehicle, for in all descriptions of the instrument we find emphasis laid on its use for determining road resistance. This application is based on the fundamental formula

$$F = \frac{W\,a}{g}$$

in which F is the accelerating force; W, the weight accelerated; a, the acceleration produced; g, the acceleration due to gravity.

In endeavoring to determine the road resistance, the car is brought up to a certain speed, the power is then shut off and the car is allowed to decelerate. In this case therefore, we have to do with a decelerating force, the traction resistance, but the formula applies the same.

The traction resistance is usually wanted in pounds per ton. Therefore, if it is found that the car decelerates at the rate of 0.8 ft. p.s./p.s. the road resistance is

$$F = \frac{2000 \times 0.8}{32.16} = 50 \text{ lb. p. ton (appr.)}$$

Principle of the Device.—A diagrammatic view of the Wimperis accelerometer is shown herewith. The principle is that of an unbalanced disk pivoted on a vertical axis. The disk is made of copper and has a circular hole cut through it to one side of the axis so as to bring the center of gravity to the opposite side of the axis. In the neutral or zero position of the disk, the line connecting the center of gravity and the axis of rotation of the disk is or should be at right angles to the direction of motion. If now motion is imparted to the instrument, that is, if it is accelerated, the heavy side of the disk will tend to lag behind. This will cause the disk to rotate around its axis against the force of a spiral spring. The force acting on the center of gravity of the disk is directly porportional to the acceleration, but the torque or turning moment due to this force will not be, for the reason that as the disk moves from its right-angled position the lever arm through which the turning force acts becomes smaller. The counteracting spring obeys a straight-line law; that is, its torsional yield is directly proportional to the torsional force impressed upon it and allowance is made for the variation in the lever arm in the scale of the instrument, the divisions of which become smaller as the distance from the zero mark increases. The instrument is of the central zero type.

Damping Means.—The reason for making the disk of copper is that some damping device has to be provided to prevent constant swinging to and fro of the disk and the pointer and a copper disk mounted between the poles of a magnet affords this damping action. The poles of the magnet come fairly close together and straddle the disk, so that the magnetic lines of force are compelled to pass through the disk, resulting in the generation of eddy currents therein which react with the magnetic field of force in such a way as to tend to prevent the motion.

Means are also provided in the instrument for making it immune to any transverse acceleration due to swaying of the car. A small spur gear is mounted on the axis of the disk and meshes with another spur gear whose shaft has a bearing supported by the frame of the instrument, and carries the indicating hand. This latter spur gear is also loaded eccentrically in such a way as to exactly balance the effects of transverse acceleration on the copper disk. The effect of transverse acceleration on the latter need be considered only when the disk is deflected from the neutral position, because, in the neutral position, this accelerating force passes through the axis of the disk and therefore cannot exert any turning influence. In the sketch Fig. 2 herewith the gear fastened to the copper disk is shown deflected from the neutral position through an angle a and the driven spur gear is therefore turned from its neutral position through the same angle, but in the opposite direction. It will readily be seen that any force F along the line of tangency of the two pitch circles will cause the two gears to turn about their respective axes, one right handedly, the other left handedly, whereas a force F' at right angles thereto (transverse acceleration) will produce no effect, as both gears will tend to turn in the same direction, which is impossible.

In this way, all transverse accelerating forces are taken care of. In order to obviate any disturbance by vertical accelerating forces it is only necessary to adjust the instrument horizontally, as then any vertical forces cannot produce motion of the disk around its axis. To this end the instrument is provided with three legs of which one is adjustable. Not only is it necessary, in order to obtain accurate results, that the instrument should be carefully adjusted in a horizontal plane, but the fore and aft axis of the industry must be in line with the direction of travel.

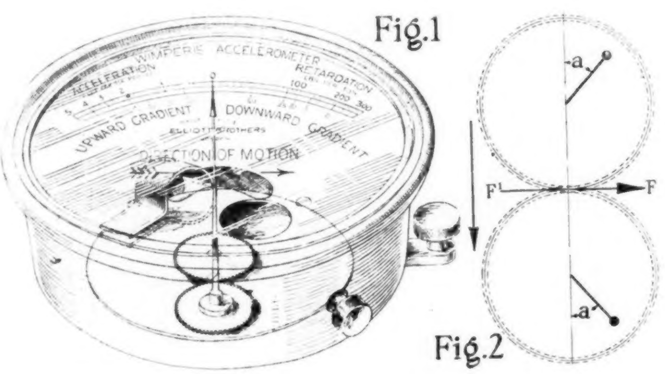


Fig. 1—Sketch of Wimperis accelerometer

Fig. 2—Principle by which accelerometer is rendered immune to effects of transverse acceleration

Method of Using Instrument.—In determining acceleration curves by means of the accelerometer the usual plan is to use both a speedometer and an accelerometer and take simultaneous readings. Unfortunately, neither the speedometer nor the accelerometer is a dead beat instrument and it is practically impossible to get accurate readings from them when the speed and acceleration are varying. This makes the accelerometer unsuitable for official tests for record or other purposes. When the factor to be measured is substantially constant it is possible to get a fairly accurate value even with a non-dead beat instrument by halving the extreme deflections of the indicator hand, but when the factor to be measured is subject to rapid variation this method fails. The chances for error are especially great in the case of a factor like acceleration, which has no inertia and can change instantly from the maximum positive to the maximum negative value.

The necessity for reading two instruments simultaneously is another objection, as it calls for two observers.

Instead of using a speedometer and an accelerometer simultaneously it is possible to use either a speedometer or an accelerometer together with a stop watch. This method, however, is equally unsatisfactory, as in addition to the difficulty of accurate readings of the speedometer or accelerometer, there is the further objection that the observation data are not in the form desired and the results have to be converted from a time-speed basis or a time-acceleration basis to a speed-acceleration basis.

Relation Between Acceleration, Distance and Time.—Acceleration in the dimensional system is represented by the expression L/T^2 and if it has a constant valve for an appreciable period it can easily be determined by simply measuring the distance moved by a body and dividing this by the square of the time in which the motion occurred, supposing the body to have started from a standstill. Unfortunately in practice such a thing as uniform acceleration is almost unknown. The acceleration may be considered constant, however, during an instant of time, and is then equal to the second differential coefficient of the distance with respect to the time—

$$a = \frac{d^2l}{dt^2}$$

It will be seen that only two fundamental quantities, length and time, have to be measured, the same as in making speed determinations, and the same apparatus can therefore be used for the purpose.

Some years ago the Society of Automotive Engineers was requested to work out a method of making acceleration measurements, and a subdivision of the Standards Committee, known as the Research Division, was entrusted with the work. This division evolved an apparatus which was entirely automatic, a drum at or near the starting line being revolved at uniform speed by an electric motor and having a time record inscribed upon its surface by means of a marker actuated by clock work. Distances of uniform length are measured off along the course over which the test is to be made, and at each point an electric switch is located which is closed by a gate secured to the front axle of the car, in passing over it. The closing of the electric circuit by one of these switches causes another marker to inscribe a record upon the drum, and in this way an accurate record of time elapsed while covering different distances from the starting point is obtained. The problem is then to convert this distance-time record into a speed-acceleration record, as it is usually the latter that is wanted. This can be done by means of a method of graphical differentiation.

Graphical Differentiation.—If observations have been taken of distances covered in various times from a standing start, we can plot the results on a coordinate diagram, plotting time along the axis of abscissas and distances along the axis of ordinates. The speed at any time is then represented by the expression dl/dt which also represents the tangent of the angle of slope. Suppose that we plot the distances to a scale of 1 in. = 100 ft. and the time to a scale of 1 in. = 2 seconds. Then, when the curve has an inclination of 45 deg., so that the value of the tangent is 1, the speed is evidently 100 ft. in 2 seconds or 50 ft. per second. For any other inclination of the curve we need only multiply this figure of 50 ft. p. s., which is determined by the scales of the diagram, by the tangent of the inclination of the curve. In this way the speed corresponding to any time can be determined and after

the figures have been obtained a speed-time curve can be plotted. Then in exactly the same way the time-acceleration curve can be obtained from the time-speed curve. The method is illustrated in Fig. 3. Here the vertical scale is 100 ft. p. in. and the horizontal scale 2 seconds to the inch, hence the speed is 50 ft. p. s. where the inclination of the curve is 45 deg. It is

$$0.577 \times 50 = 28.85 \text{ ft. p.s.}$$

where the inclination is 30 degrees, and

$$1.732 \times 50 = 86.6 \text{ ft. p.s.}$$

where the inclination is 60 degrees.

By means of a table of natural trigonometric functions the speed corresponding to any point of the time-distance curve can then be found.

Another Method.—Another method of graphical differentiation is as follows: The curve which is to be differentiated is first drawn and the same curve is then plotted a second time a small distance to the right of the first. Then the vertical distance between the two curves at any point is an ordinate of the differential curve for an abscissa smaller than the abscissa of the point by one-half the horizontal displacement of the two curves. In Fig. 4, let *ABC* be the time-distance curve and *A'B'C'* a like curve shifted to the right a distance *AA'*. Then the vertical distance *aa'* between the two curves taken at the point *b* of the time scale, is equal to the ordinate of the time-speed curve at a point *c* of the time scale, the distance *bc* being one-half the distance *AA'*. Rather than shifting all the ordinates in this way, the time scale for the time-speed curve could be shifted to the right a distance *AA'*/2. This method is quite handy, and the only objection to it is that when two differentiations are performed in succession, as is necessary in this case, the ordinates become very small. This can be remedied by multiplying the ordinates of the derived curves by a constant factor so as to make the curves more legible and better utilize the available space on the curve sheet.

Predetermination of Accelerating Power.—It is sometimes desirable to determine approximately the acceleration of a car not yet built, and if the horsepower curve of the engine is known and the weight and air resistance area can be estimated, this can be done with a fair degree of accuracy.

In Fig. 5 is shown a horsepower curve of a 6-cylinder 3½ x 5¼ in. engine. This is fitted to a car with a rear axle reduction of 4.45, rear wheels 34 in. in diameter and weighing, with one passenger, about 3700 lb. The low gear reduction in the transmission is about 3 to 1 and the intermediate 1.73 to 1. We may assume that the equivalent wind resistance area is 16 sq. ft.

The power requirements of the car at various speeds may then be found from the equation

$$HP = \frac{Wv}{15,000} + \frac{Av^3}{80,000}$$

where W is the weight of the car and load, v the speed in miles per hour, and A the projected area in square feet.

The air resistance factor (the second term of the right-hand side of the equation) is always more or less uncertain. It is known that the air resistance depends not only upon the projected area of the car, but also upon the general form of the body, a streamline form, for instance, encountering much less resistance than one more blunt. However, at speeds up to 30 m.p.h. this uncertainty does not greatly affect the value obtained for the acceleration.

By means of the above equation we can draw a curve of horsepower required at different speeds. Some allowance must be made for loss in the transmission, besides which the engine will hardly produce as much power in the car as on the test stand when it is tuned to the very highest pitch. If we figure on 85 per cent of the test horsepower on the rear wheels at any given speed, we will not be far wrong.

Let

N = revolutions per minute of the engine
S = car speed in miles per hour
r = high gear reduction ratio
D = wheel diameter in inches
F = accelerating force acting on car (lb.)
a = acceleration in ft. p.s. p.s.
W = weight of car and load ,

then

$$N = \frac{336\, rS}{D}$$

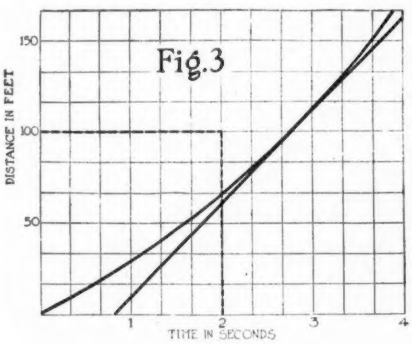
$$F = \frac{375\, HP}{S}$$

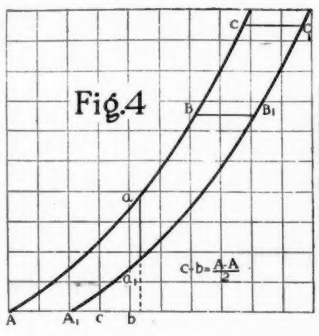
and

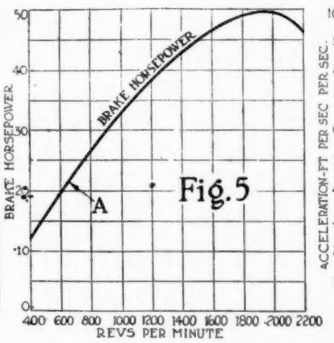
$$a = \frac{32.16\, F}{W}$$

On the intermediate and low gears, an efficiency of 75 per cent can be figured with. That is, we can safely assume that 75 per cent of the horsepower shown by the curve for any engine speed will be available at the wheel rims. This car has an intermediate gear reduction in the gear box of 1.73 to 1. The accelerations on intermediate gear under full throttle at different speeds are calculated as shown in Table 2.

The figures for low gear acceleration are given in Table







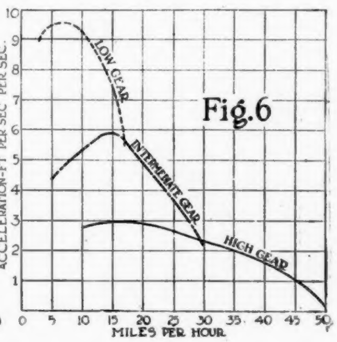


Fig. 3—Method of graphical differentiation. Fig. 4—Another method of graphical differentiation. Fig. 5—Horsepower curve of car "A." This car is fitted with a six-cylinder 3½ x 5¼ engine geared 4.45 to 1. It has 34-in. wheels and with one passenger weighs about 3675 lb. Fig. 6—Acceleration curves for all these gears of car "A"

TABLE I—HIGH GEAR ACCELERATION OF CAR "A" AT DIFFERENT ROAD SPEEDS

Road speed, m.p.h	10	20	30	40	50
Power to overcome road resistance	2.47	4.94	7.41	9.88	12.34
Power to overcome air resistance	0.20	1.60	5.40	12.80	25.00
Total power for propulsion	2.67	6.54	12.81	22.68	37.34
Corresponding engine speed	440	880	1310	1760	2200
Power available at wheel rims	11.45	24.60	34.80	41.60	39.40
Power available for acceleration	8.78	18.06	22.00	18.92	2.06
Accelerating force, lb	328.00	339.00	275.00	177.5	15.5
Acceleration, ft. p. s. p. s	2.84	2.93	2.38	1.59	0.13

TABLE II—INTERMEDIATE GEAR ACCELERATION OF CAR "A" AT DIFFERENT ROAD SPEEDS

Road speed, m.p.h	5	10	15	20	25	30
Power to overcome road resistance	1.23	2.47	3.70	4.94	6.17	7.41
Power to overcome air resistance	0.02	0.20	0.67	1.60	3.12	5.40
Total power for propulsion	1.25	2.67	4.37	6.54	9.29	12.81
Corresponding engine speed	382	764	1146	1528	1910	2292
Power available at wheel rims	8.04	18.75	27.75	34.30	37.50	33.00
Power available for acceleration	6.79	16.08	23.38	27.76	28.21	20.20
Accelerating force, lb	508	615	684	520	423	252
Acceleration, ft. p. s. p. s	4.40	5.32	5.92	4.93	3.66	2.18

TABLE III—LOW GEAR ACCELERATION OF CAR "A" AT DIFFERENT SPEEDS

Road speed, m.p.h	3	5	10	15	17
Power to overcome road resistance	0.74	1.23	2.47	3.70	4.20
Power to overcome air resistance	.....	0.02	0.20	0.67	0.98
Total power for propulsion	0.74	1.25	2.67	4.37	5.18
Corresponding engine speed	395	660	1320	1975	2240
Power available at wheel rim	9.00	15.75	31.20	37.50	34.10
Power available for acceleration	8.26	14.50	28.53	33.13	28.92
Accelerating force, lb	1030	1088	1070	844	638
Acceleration, ft. p. s. p. s	8.90	9.42	9.27	7.31	5.50

3, the efficiency of transmission being taken as 75 per cent here too.

Acceleration Curves.—Acceleration curves for all three gears are plotted in Fig 6. From this it will be seen that for maximum acceleration the gear should be shifted from low to intermediate when a speed of about 17 m.p.h. has been attained, and from intermediate to high when the speed has reached about 28 m.p.h.

In Fig. 7 are shown high gear acceleration curves for two other cars, *B* and *C* for the sake of comparison. All necessary data of the two cars are given in the captions.

In the acceleration of the car by means of the gearset there are a number of intermediate stages during which the car first decelerates and then is accelerated—sometimes exceedingly rapidly—by energy being withdrawn from the engine flywheel. The start from a standstill is made largely by means of energy withdrawn from the flywheel. The engine is brought up to a moderate speed, when the clutch is let in, and as the clutch takes hold the engine speed decreases and the car speed increases until the two correspond and slipping of the clutch ceases. At this moment the engine will usually be moving at a pretty low speed, especially if it is desired to make a quick getaway and the driver lets the clutch in quickly.

Most Advantageous Speed for Shifting.—From the curves, Fig. 6, it would seem to be most advantageous to accelerate on the low gear up to a speed of about 17 m.p.h. In practice, however, the change to intermediate would be made somewhat earlier, first because the engine would have to attain a very high speed to give this speed on the low gear, at which it would probably be somewhat noisy, and, second, because when the intermediate is put in mesh and the clutch is let in quickly, owing to the much greater relative speed of the engine as compared with the car, the acceleration immediately after the clutch is fully engaged will be much greater than that shown by the intermediate gear acceleration curve, due to energy supplied by the flywheel. During the moment that the clutch is entirely disengaged the car speed, of course, is decreasing; in other words, there is negative acceleration or retardation.

From tests made with recording accelerometers it appears that this period of negative acceleration is of the order of one second. When the clutch takes hold again the acceleration changes from negative to positive and, as stated, it may even attain a value higher than that corresponding to the particular car speed and gear. This produces that unpleasant jerk sometimes experienced when shifting gear and jamming in the clutch. With good driving the acceleration on engaging a new gear should not materially exceed that which corresponds to the junction of the acceleration curves for low and second gears.

Time-Speed Curves.—From the acceleration curves it is possible to draw a time-speed curve, though those portions of the curve during which the clutch slips are somewhat uncertain. At a speed of 3 m.p.h. the low gear is capable of giving an acceleration of about 9 ft. p.s. p.s. and as 3 m.p.h. corresponds with 4.4 ft. p.s., with constant acceleration it would take ½ second to attain this speed, at which we may assume the clutch to take firm hold. From this point on the acceleration follows the curve for the low gear, which we may assume to remain in mesh until a speed of 15 m.p.h. is reached. In order to determine the time consumed in accelerating the car from 3 to 15 m.p.h. we divide this portion of the acceleration curve into a number of sections, during each of which the acceleration remains substantially constant. We then take the mean acceleration for that section and divide the increase in speed occurring during that section, by the rate of acceleration, which gives the time corresponding to that section. Thus, while the speed increases from 3 to 5 m.p.h., the average rate of acceleration is 9.1 ft. p.s. p.s. and the increase in speed is 2 m.p.h. or

$$2 \times 1.466 = 2.932 \text{ ft. p.s.}$$

Hence the time occupied in getting from a speed of 3 m.p.h. to a speed of 5 m.p.h. is

$$2.932 \div 9.1 = 0.32 \text{ second.}$$

In passing from 5 to 10 m.p.h. the acceleration remains substantially constant at 9.4 ft. p.s. p.s. and the time consumed is

$$\frac{5 \times 1.466}{9.4} = 0.78 \text{ second.}$$

From 10 to 12.5 m.p.h. the average acceleration is 8.8 ft. p.s. p. s. and the time

$$\frac{2.5 \times 1.466}{8.8} = 0.42 \text{ second.}$$

From 12.5 to 15 m.p.h. the average acceleration is 7 ft. p.s. p.s. and the time

$$\frac{2.5 \times 1.466}{7.8} = 0.47 \text{ second.}$$

Hence in the period of $0.30 + 0.78 + 0.42 = 2.29$ seconds the car is speeded up to 15 miles per hour. Here an in-

terruption of about 1 second occurs in the acceleration of the vehicle. For about one-half this time, while the clutch is out, the car will be drifting. Since the total traction resistance at 15 m.p.h. corresponds to a power expenditure of 4.37 hp. and the excess available power of 33.13 hp. produces an acceleration of 7.31 ft. p.s. p.s., the retardation will be at the rate of

$$\frac{4.37 \times 7.31}{33.13} = 0.96 \text{ ft. p.s. p.s.} = \frac{0.96}{1.466} = 0.66 \text{ m.p.h. p.s.}$$

In one-half a second the speed will therefore drop from 15 m.p.h. to 14.67 m.p.h. This loss in speed will be made good again during the next half second while the clutch is being let in, so that when the intermediate gear gets a firm hold the car is traveling at 15 m.p.h. On the intermediate gear the car may be accelerated to 25 m.p.h. The times required for different fractions of the acceleration from 15 to 25 m.p.h. are as follows:

15 — 17.5 m.p.h.—

$$\frac{2.5 \times 1.466}{5.65} = 0.65 \text{ sec.}$$

17.5 — 20 m.p.h.—

$$\frac{2.5 \times 1.466}{5.2} = 0.70 \text{ sec.}$$

20 — 22.5 m.p.h.—

$$\frac{2.5 \times 1.466}{4.6} = 0.80 \text{ sec.}$$

22.5 — 25 m.p.h.

$$\frac{2.5 \times 1.466}{4} = 0.92 \text{ sec.}$$

Hence the total time required to accelerate from 15 to 25 m.p.h. is $1.00 + 0.65 + 0.70 + 0.80 + 0.92 = 4.07$ sec.

Now the clutch is withdrawn again for half a second. While the clutch is out the car will decelerate again, this time by

$$\frac{9.29 \times 3.66}{2 \times 28.21} = 0.6 \text{ m.p.h.}$$

so that the speed drops to 24.4 m.p.h. but this loss is gained again while the clutch is taking hold, so that when the slipping ceases the speed is again 25 m.p.h.

From here on we accelerate on the high gear and the times required for different speed increases are as follows:

25 — 30 m.p.h.—

$$\frac{5 \times 1.466}{2.5} = 2.93 \text{ sec.}$$

30 — 35 m.p.h.—

$$\frac{5 \times 1.466}{2.2} = 3.34 \text{ sec.}$$

35 — 40 m.p.h.—

$$\frac{5 \times 1.466}{1.8} = 4.08 \text{ sec.}$$

40 — 45 m.p.h.—

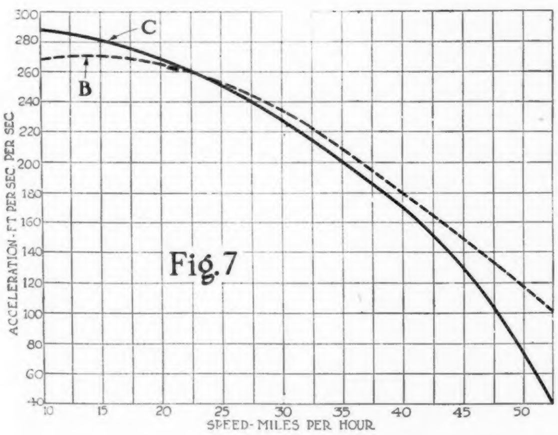
$$\frac{5 \times 1.466}{1.25} = 5.86 \text{ sec.}$$

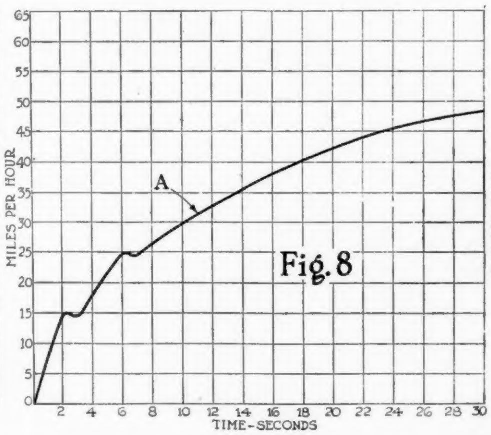
45 — 50 m.p.h.—

$$\frac{5 \times 1.466}{0.5} = 14.66 \text{ sec.}$$

Acceleration in accordance with the above figures may be all right in record trials or stunts, but is entirely too fast for comfort. In regular driving the throttle is practically never fully opened when accelerating on first and second gears. From a regular driving standpoint the acceleration on high gear from the lowest speed at which the car will run smoothly on that gear—say 10 m.p.h.—is of most interest. A speed-time curve for the "A" car covering acceleration on high gear from 10 to 40 m.p.h. is shown in Fig. 9.

IN the discussion of his paper on Motor Design read before the Minneapolis S. A. E. Section, C. W. Pendock of the Le Roi Motor Co. said that he knew of some tests where aluminum connecting rods were used without the addition of any bearings, no forged bearings of any description and no bushings, and the results were very favorable indeed. In one instance a four-cylinder engine with two aluminum and two steel rods was run, which gives a very bad condition because it throws the engine so badly out of balance, by reason of the fact that the two aluminum rods are of only about one-half the weight of the steel rods, and they stood up very well.





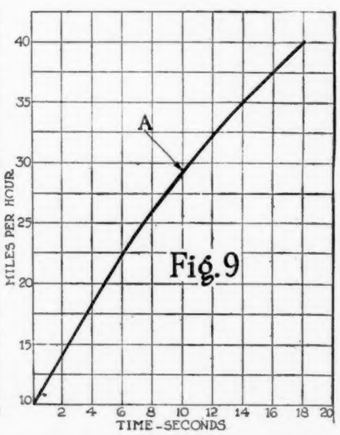


Fig. 7—High gear acceleration curves for cars "B" and "C." Car "B" with one passenger weighs 3825 lb., has a six-cylinder 3¾ x 5 in. engine geared 3.88 to 1 and is equipped with 34-in. wheels. Car "C" weighs 3150 lb. and has a six-cylinder 3¼ x 4½ in. engine geared 4.75 to 1, the wheels being 32 in. in diameter. Fig. 8—Time-speed curve of "A" car when accelerated from a stand-still on the three gears successively. Fig. 9—Time-speed curve of "A" car covering high-gear acceleration from 10 to 40 m.p.h.

Designing a More Efficient Intake Manifold

Delivery of the fuel from the carbureter to the combustion space has not received a full measure of consideration from engineers and designers. This article is a brief analysis of the problems involved.

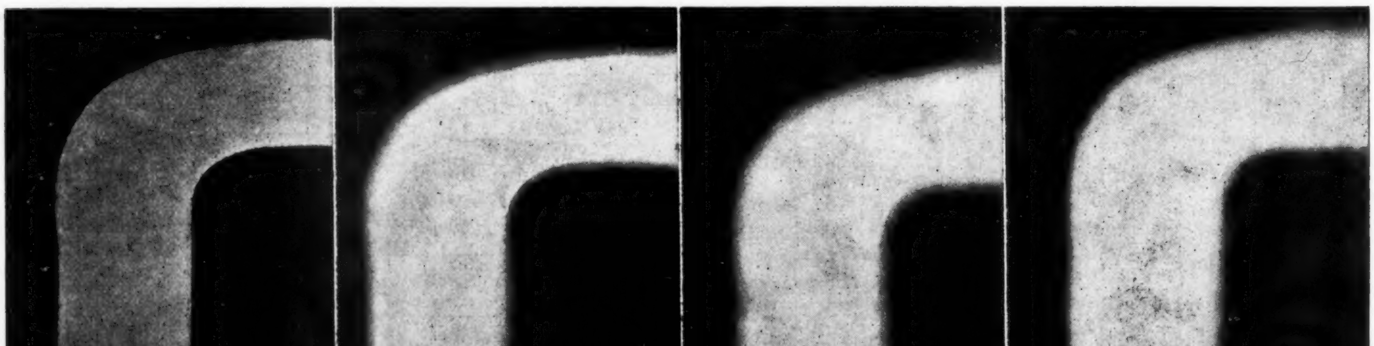
By W. D. Bell

WITH automotive fuels of the grade commercially available to-day, the best that can be expected of a carbureter is that it will produce an accurately proportioned mixture of the air with a more or less finely divided spray of the liquid. The actual combination of the two fluids to form a combustible gas is an entirely different process which takes place in the intake passages and the cylinder and which continues in most cases during the entire combustion period. This fact has a more direct bearing on the problem of successfully burning the low grades of fuel than is generally realized and it is the purpose of this paper to present graphically the effect of intake manifold design upon the operation of the engine.

It is obvious that a mechanical mixture consisting of drops of liquid suspended or floating in a stream of air is very unstable. Newton's First Law of Motion tells us that "Every body continues in its state of rest or of uniform *motion in a straight line* unless it be compelled by impressed force to change that state." A body of small mass, such as a drop of fuel in a stream of air passing through an intake manifold has a high velocity relative to the manifold walls but practically zero relative to the air immediately around it. Under such circumstances the "impressed force" tending to cause a change of direction of a particle is very slight and unless all bends are of very wide radius and carefully planned the drops will invariably strike the wall of the manifold.

In order to find out exactly what happens to the mixture while on its way to the engine cylinder, the writer constructed manifolds and sections of manifolds having glass sides so that the flow of fuel might be observed and to learn, if possible, to just what extent precipitation was apt to take place at the bends. The results of these observations were interesting, especially since, after considerable experimenting, a method was devised whereby they could be recorded photographically. A number of photographs were made of various manifold sections and at different flow rates. All of the photographs disclose the same tendencies and the same characteristics, the four which accompany this paper being selected from the number because they represent a typical case found on almost every engine and because they seemed most suitable for reproduction by the half-tone process.

The pictures are really shadows cast upon a plate by the particles of fuel when illuminated by electrical discharges of short duration. In making photograph No. 1 the exposure was so short that the motion of the particles was "stopped" completely, each globule being recorded as a mere dot on the negative. In Nos. 2, 3 and 4 the exposures were lengthened somewhat in order that the path of the particles might be determined and their velocity estimated by the extent of their movement during the interval of the exposure. It is interesting to notice that the larger drops refracted the light to such an extent that they are recorded as bright points or lines of light instead of dark shadows and that the presence of a film of liquid upon the manifold walls perpendicular to the photographic plate is indicated by a slight diffusion of the light and the consequent blurring of the outlines of the manifold. All of the pictures reproduced herewith were made of mixtures somewhat richer than those ordinarily used, in order to get a clearer and more distinct image.



No. 1 No. 2 No. 3 No. 4

Shadows cast by particles of fuel when illuminated by electrical discharges

Photograph No. 2 indicates a low velocity and a heavy mixture, a condition similar to that of a motor running slowly with wide open throttle. The fuel particles are bunched in the vertical pipe and in the outer portion of the elbow on the down-stream side, a condition very apt to cause "loading."

Photograph No. 3 is a higher velocity stream and the effects of centrifugal action are very apparent. The blurring of the outlines of the manifold is due, as stated before, to the refraction of light by the film of precipitated liquid, and the thickness of the film may be roughly estimated by the amount of the blurring.

Photograph No. 4 is a still higher velocity, and the effect of centrifugal action is more pronounced than in any of the others. There is a distinct accumulation or wave of liquid fuel on the outer wall of the elbow at the point where the rapidly moving particles are projected in their unsuccessful attempt to negotiate the bend, while a film of precipitated fuel is evident on *both* sides of the manifold.

The effect of the separating action on the character of the mixture is strikingly shown in all of the photographs. It will be noticed that in the straight portion of the passage the mixture is not at all uniform, a considerable part of the fuel being in the form of a coarse spray. At the elbow the heavier drops are almost completely separated by the centrifugal action, while the lighter particles remain in suspension and the mixture as it leaves the elbow is fairly homogeneous. This action is due to exactly the same natural laws made use of in the centrifugal air cleaners found on some tractors. It is certainly an anomalous state of affairs to place on one side of a carbureter a centrifugal device for ridding the air stream of dust and then, after exercising every bit of ingenuity possible to produce a correct mixture, subject it to centrifugal forces even more violent and still expect it to reach the cylinder safely. When fuel has once accumulated upon the walls of the manifold it is obvious that it can only reach the combustion space by being drawn along by the friction of the air stream, or by being evaporated, either mechanically or naturally.

The popular demand for high volumetric efficiency has made it necessary to keep the air charge at a temperature considerably below the dew-point of the low grade fuel vapor, and in this way has brought about the use of "wet" mixtures. Under such conditions, the film of precipitated fuel constitutes a very real problem. Any fuel which strikes the walls of the manifold or of the cylinder is almost sure to remain there and not only resists combustion but is also directly responsible for most of the troubles from carbon deposits and faulty lubrication. On the other hand, the success attained by engines of the fuel injection type proves conclusively that if properly atomized and uniformly distributed through the air charge it is possible to burn almost any grade of hydrocarbon efficiently.

This matter of delivering the atomized fuel from the carbureter to the combustion space is important, deserving much more attention than is usually given it. Complete vaporization is not a prerequisite to combustion but the fuel must be finely pulverized and intimately mixed with the air charge. Study of the photographs will make it clear that wherever a bend must be negotiated by the mixture in its passage through the manifold centrifugal force will be set up and there is apparently no way of preventing precipitation. It would seem that as long as the manifold must be retained in anything like its present form the most logical solution of the problem would be to heat highly a limited section of the wall at the outlet of each elbow, so that any precipitated fuel will be boiled off and again enter the air stream. It is not sufficient to hot-spot a single elbow, nor advisable to heat entire manifold.

Quite a lot has already been accomplished along this line, but there is much yet to be done. A little intelligent co-operation between carbureter makers and engine designers should make it possible to evolve a system combining the merits of the carbureter and the injection engine, in which case the fuel problem in its present form would cease to exist.

Association of German Industries

THE Association of German Industries was formed in 1919, and has for its purpose the representation and development of the German industries, the determination of a uniform policy of each separate trade and common action in dealing with labor questions. It is composed of a council and directorate, which latter consists of at least thirty and not more than sixty members, who either occupy or have occupied a leading position in an industrial undertaking or have held a position on a board of directorate.

The council is empowered to nominate ten additional members to the directorate. The council, which consists of from seven to fifteen members, is elected from the members of the directorate. The chairman of the council is Dr. Ingenieur Kurt Sorge, Berlin; the first representative of the chairman is Abraham Frowein, Elberfeld; the second representative of the chairman is C. F. v. Siemens, Berlin-Siemensstadt; and the managing director, Privy Councillor Dr. W. Simons.

A main committee has been nominated, states the *Deutsche Bergwerks Zeitung*. It consists of representatives of each of the twenty-five industrial branches into which the association is graded. The total number of representatives is 140, distributed as follows: Mining, 15; steel works, 10; metal works and metal half-finished products, 3; machines, 5; railway cars, 2; hardware, 4; iron and steel goods, 5; electro-technical precision instruments and optics, 3; boilers and fittings, 2; automobiles and bicycles, 3; timber, 5; leather and leather manufactures, 4; stone and earth, 6; building, 3; pottery, 3; glass, 4; chemical, 10; oils and fats, 3; paper, 8; textiles, 23; clothing, 3; breweries and flour and malt mills, 3; sugar and foodstuffs, 4; provisions, 3; shipping and transport, 3.

In addition to the 140 members of the industrial groups, the main committee is further composed of ten representatives of agricultural associations, ten representatives of home industries elected by members meeting at the suggestion of the directorate, and ten representatives of members of the association nominated at the suggestion of the directorate.

The main committee nominates special committees and elects from year to year a committee of investigation. The following special committees have been formed: (1) Committee for the carrying out of the Articles of the Treaty of Peace; (2) tax committee; (3) economic committee; (4) social-political committee; (5) press committee; (6) committee of investigation.

A new issue of the Auto Electrician's Guide, the book of wiring diagrams published by the Michigan State Auto School, is now ready and contains over 900 diagrams, including those of 1920 models.

Comparison of Hecter Fuel with Export Aviation Gasoline

This fuel, composed of approximately 30 per cent benzol and 70 per cent cyclohexane, was thoroughly tested by the Bureau of Standards in a Liberty engine under varying altitude conditions, compared with standard fuel. Results are summarized in this report which has just been made public.

By H. C. Dickinson, V. R. Gage and S. W. Sparrow*

AVIATION engine developments for attaining higher power at altitude are following two principal lines, supercharging and increase in compression ratio. For the latter, fuels have been demanded which are capable of operating under compressions too high for gasoline. The U. S. Bureau of Mines in a comprehensive investigation of fuels for internal combustion engines, found cyclohexane mixtures to possess this property which led them to develop the "Hecter" fuels. The ability of this fuel to withstand high compression without knock was demonstrated in an experimental engine at ground level and its general usability was proved by actual flight tests. It was accordingly submitted to the Bureau of Standards for comparison with X gasoline as to the relative power producing ability and fuel consumption of the two fuels when used in an engine with an extremely high compression ratio (7.2).

The gasoline used in these tests was the standard reference fuel of this laboratory (known as "X" gasoline), prepared for the Bureau of Standards by the Atlantic Refining Co. from Pennsylvania crude oil. It complies with specification No. 3512 of the Bureau of Aircraft Production for Export Aviation gasoline for the use of the American Expeditionary Forces in 1918. The Hecter fuel supplied for these tests was a mixture of approximately 30 per cent benzol, 70 per cent cyclohexane by volume. The properties of both fuels are shown in Fig. 1.

The fuels were tested in a twelve-cylinder Liberty airplane engine equipped with special pistons giving a compression ratio of 7.2. This ratio gave a measured compression pressure of 170 lb. per sq. in. gage. The engine was mounted in the altitude chamber of the Bureau of Standards automotive power plants laboratory where controlled conditions of air pressure and temperature approximate those prevailing at altitudes up to 30,000 ft.

In these tests two fuel tanks were used, one containing the X gasoline, the other Hecter. The engine was started on X gasoline and the desired conditions of speed and altitude reached with a comparatively rich mixture. The maximum dynamometer (engine) torque having been attained, observations of torque were continued while the rate of gasoline supply was gradually reduced. The mixture was leaned until the torque fell off considerably, then gradually enriched just enough to regain the maximum torque which had previously been noted. Readings were then taken of the various temperatures, pressures, torque, rates of flow, speed, etc. The fuel supply from the X tank was then cut off and Hecter supplied to the carbureter. After adjusting the carbureter for maximum torque with the least expenditure of fuel as described for X, readings of test data again were made. By changing from one fuel to another in this manner, it is possible effectually to eliminate from the comparison of the fuels any effect due to changes in the condition of the engine.

The test results are shown graphically in the accompanying curves. Fig. 2 shows the brake horsepower for Hecter, Fig. 3, the brake horsepower for X gasoline, Fig. 4, the fuel consumption for both fuels, Fig. 5 is the heat balance, and Fig. 6, the performance comparison on a percentage basis, all results being plotted against altitude.

In the heat distribution, the percentage of heat in brake horsepower, exhaust, and jacket is measured and the "residual" heat is the term given to the difference between the sum of these values and 100 per cent. This residual heat, therefore, includes the unburned fuel in the exhaust and the so-called radiation losses, less the heat supplied by the combustion of the lubricating oil. The heat supplied is computed from the higher heating value of the fuel and the exhaust heat is measured by "exhaust calorimeter"

Abstract from N. A. C. A. Report No. 90 and A. P. P. Report No. 67.

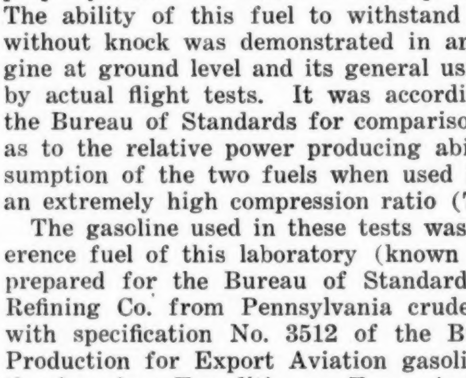


Fig. 1

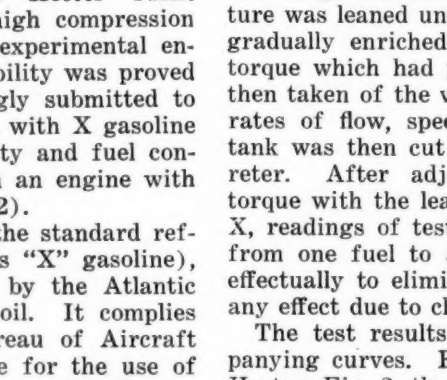


Fig. 2

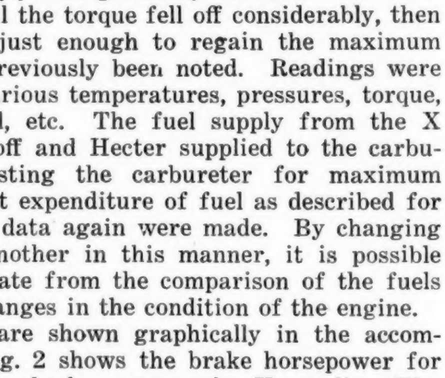


Fig. 3

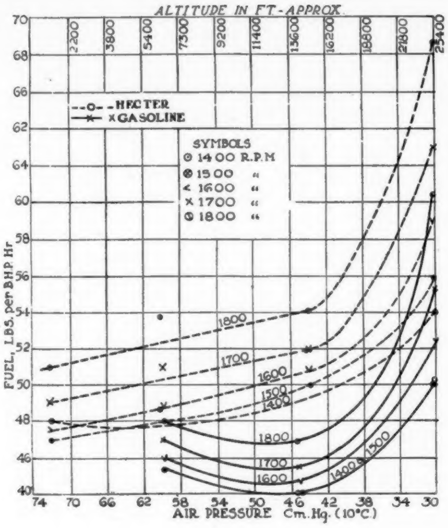


Fig. 4

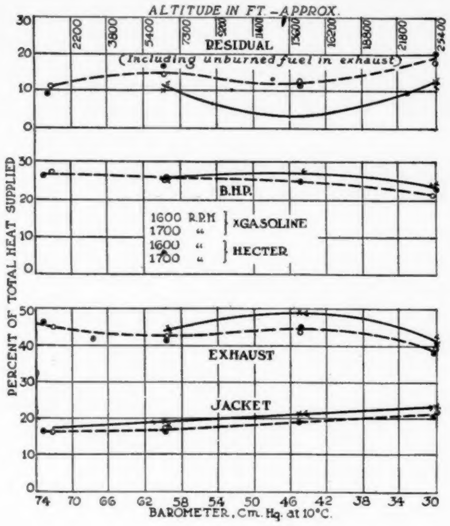


Fig. **5**

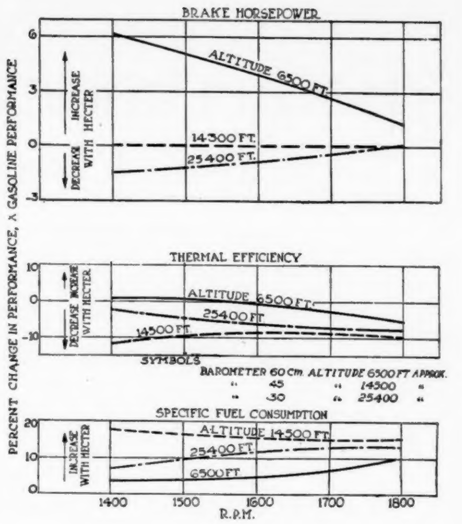


Fig. 6

methods. The residual heat when using Hecter is always more than when using X gasoline, the exhaust and jacket losses and the brake thermal efficiency are always less. The interpretation is that less of the heat energy of Hecter is liberated in the cylinder and more of the fuel is exhausted unburned. These curves should not be construed as showing the exact quantitative effect of altitude changes alone upon heat distribution, being considerably influenced by the carbureter adjustment. The reverse curvature of the exhaust and residual lines, indicating a more complete burning of X gasoline at 12,000 ft. is, however, a tendency noted on other tests.

Conclusions: 1. For flight at low altitudes Hecter fuel showed slight advantages in comparison with gasoline by affording a small increase of power over and above that necessary to offset the disadvantage of increased fuel consumption. The usual ratio of fuel weight to plane weight is of the order of 1 to 7, so that for full throttle flying an increased fuel consumption of 7 per cent balances an increase of 1 per cent in power developed. The test at 6500 ft. altitude showed that Hecter fuel developed slightly more power than X gasoline, the maximum advantage being 7 per cent and the average for all speeds 4 per cent, whereas the increase in fuel consumption averaged 5 per cent or 6 per cent. Since at 14,000 ft. and 25,000 ft. no appreciable difference in power was obtained, whereas the fuel consumption of Hecter was greater to the extent of 15 per cent by weight, the advantage lies with X gasoline.

2. The large difference in densities of Hecter fuel and X gasoline make the fuel comparisons by weight and by volume read quite differently, and care must be exercised to distinguish them. Upon reducing lb. per brake horsepower hour to pints per brake horsepower hour it is found that Hecter consumption is the less by volume at ground, and about equal, to that of X gasoline at 25,000 ft.

3. One gallon of Hecter contains nearly 9 per cent more heat units than a gallon of X gasoline, and the brake thermal efficiency of this engine using Hecter is about the same per cent less than when using X gasoline. Thus the same tank full of either fuel would supply a plane with about the same available energy. Any part of a flight at very low altitude might be accomplished at a slightly higher plane speed with the Hecter than with gasoline, as a consequence of the power characteristics described above.

4. It has been claimed that a high compression engine has a greater factor of safety when operated with Hecter fuel than with gasoline. The engine was not operated for a sufficient period of time to ascertain whether engine deterioration was more rapid with the 7.2 compression ratio than would be expected from experience with the 5.6 compression ratio. Consequently no comparison can be made of the effect of compression on fuels upon engine deterioration.

5. However, since it is not generally considered advisable to operate an engine of this type with gasoline at a higher compression ratio than 5.6, it is of interest to compare the performance of a Liberty twelve-cylinder aviation engine of 5.6 compression ratio using gasoline with the performance of the same type of engine with 7.2 compression ratio using Hecter. Previous tests with this type of engine have shown that this change in compression produces about 10 per cent increase in power with about the same percentage decrease in weight of fuel consumed per unit power. This change would be expected from a comparison of the "air standard" efficiencies. From these data it is concluded that Hecter in a 7.2 compression ratio engine would produce about 10 per cent more power than would X gasoline in a 5.6 compression ratio, while using the same weight of fuel per unit power as for X gasoline in the lower compression.

Test for Uniformity of Steel Bars

At the request of a subcommittee of the National Research Council, 54 steel bars 1 in. in diameter and approximately 12 ft. long have been examined for magnetic uniformity by the Bureau of Standards. These bars are intended for experiments on the heat treatment of carbon steel and it was desired to make sure that there were no chemical segregations or other inhomogeneities existing in the bars.

The determination was made by passing the test specimens at a uniform rate through a magnetizing solenoid (energized by a direct current) and noting the deflection of a sensitive electrical instrument connected to a system of test coils located within the magnetizing solenoid and surrounding the specimen. If the material is uniform, there is no deflection of the instrument, while any departure from uniformity is indicated by corresponding deflections. It is possible to make a photographic record of these deflections.

LIEUTENANT COMMANDER LIND, formerly a teacher at the United States Naval Academy, has written a practical treatise on Internal-Combustion Engines. The treatise is intended as a textbook for engineering classes and also as a practical reference book covering the essential features of the various types of engines.

Experimental Separation of Engine Losses Develops Interesting Results

A series of tests was conducted recently to determine the mechanical losses in various engine parts. The largest single item was found to be due to the piston ring. These tests are described in the following article.

By J. Edward Schipper

TO determine the mechanical losses due to piston rings, a series of interesting tests have been made by the Engineering Department of the Aircraft Service, McCook Field, Dayton, Ohio. In determining the relative magnitude of the mechanical losses in the various parts of the single-cylinder Liberty engine, the object aimed at was the discovery of methods of reducing these losses in the Liberty multi-cylinder engine. While these tests are applicable specifically to the Liberty engine, they are, nevertheless, of broad and general interest, as the data derived may be utilized in connection with any type of internal-combustion engine.

The results of this test show rather clearly the relative magnitudes of the various mechanical losses in the engine. The total mechanical loss varies from 13 per cent of the indicated horsepower at 1000 r.p.m. to 21 per cent at 2000 r.p.m., being 17.9 per cent at normal speed and power.

The largest single item under mechanical losses is the loss due to the piston, piston rings and connecting-rod. This is approximately 49 per cent of the total friction loss at normal speed, or 8.7 per cent of the indicated horsepower. It is probable that the connecting-rod itself contributes only a small part to this loss, although no actual data was obtained to substantiate this. The piston and its rings, therefore, seem to be the most important factor in producing this loss and should be carefully considered in attempting to reduce engine friction. The top ring accounts for a loss of from 0.2 to 0.3 of a horsepower at several points in the speed range, while at other points no loss could be measured. It is probable that the loss due to the top ring alone is too small to be accurately measured with the apparatus at hand, and the determinations of friction loss due to two rings are therefore considerably more reliable. This varied from 0.2 to 0.5 horsepower, being 0.4 horsepower at normal speed, or over 5 per cent of the total mechanical loss. While this is not a very large item in the single-cylinder engine, it would be considerably magnified in the multi-cylinder types, and this point is therefore worthy of consideration. It is noticeable, however, that the removal of each ring pro-

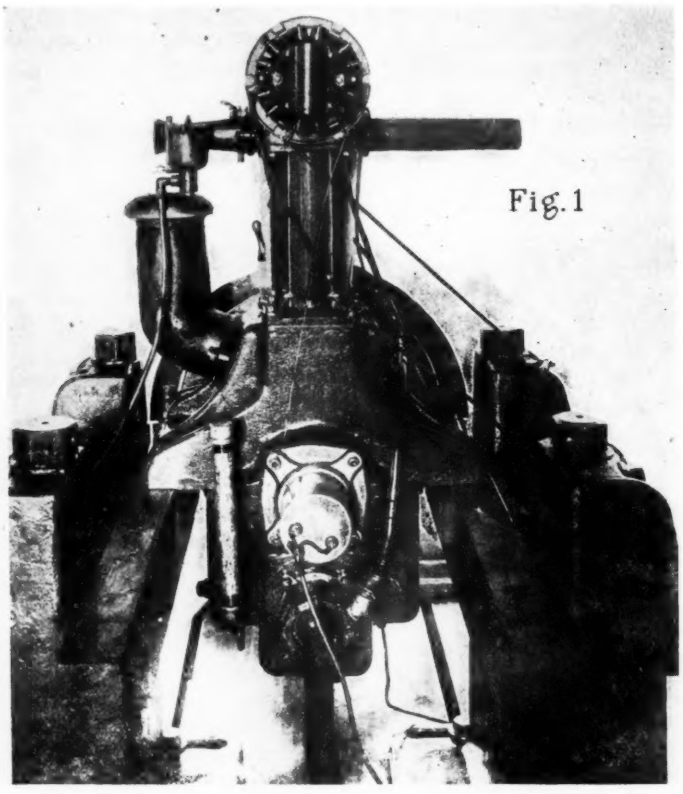


Fig. 1—Single-cylinder Liberty engine, distributer end

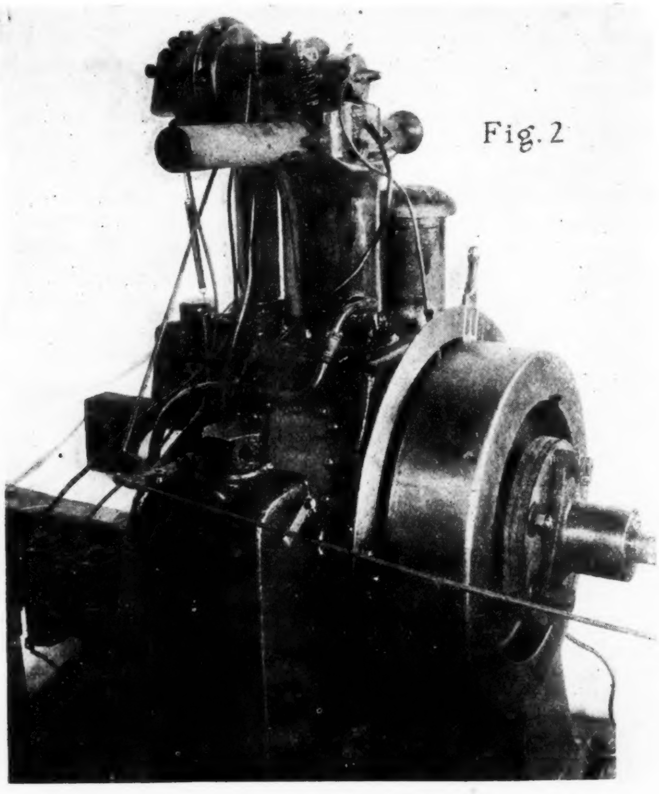


Fig 2—Single-cylinder Liberty engine, flywheel end

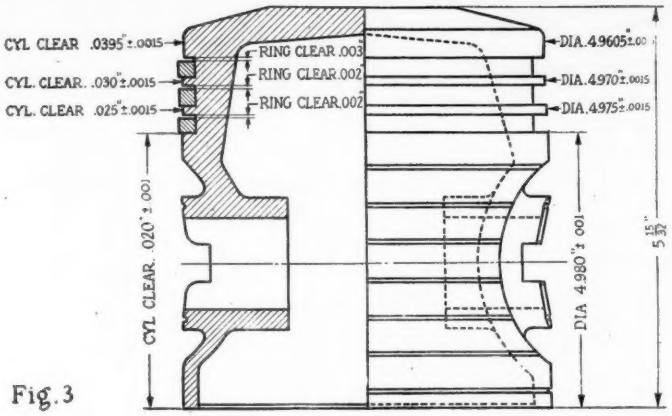


Fig. 3

Fig. 3—Drawing showing clearances of piston and piston rings of standard Liberty engines

duces a marked drop in compression pressure (see Fig. 8), and it is therefore open to question whether the reduction in friction is not offset by leakage past the piston.

The pumping loss constitutes about 30 per cent of the total mechanical loss at normal speed, or over 5 per cent of the indicated horsepower. This can only be reduced by decreasing the friction of the gas passages and valves both for inlet and exhaust. This does not offer very great possibilities, as the present engines are nearly as good as possible in this respect. That portion of the pumping loss due to the heat lost to the cylinder walls is inherent in the cycle and cannot be reduced or eliminated. Just how large a part this plays in the total pumping loss was not determined by these tests.

The crankshaft, camshaft and auxiliaries constitute the third largest factor in producing mechanical loss. This was 1.6 hp. at 1700 r.p.m., which is 21 per cent of the total mechanical loss and 3.8 per cent of the indicated horsepower. While this does not constitute as important an item as the piston friction, it is, nevertheless, worthy of consideration. By careful design of bearings, careful balancing and accurate proportioning of auxiliary units, the friction losses in these parts might be considerably reduced. In this connection it must be remembered that the losses due to the crankshaft, camshaft and auxiliaries in the single-cylinder engine are probably much greater in proportion to the power developed than in the multi-cylinder models. This is to be expected, as when the number of cylinders is increased a corresponding increase in the size of the crankshaft, accessory drives and auxiliary units is not required. It is, therefore, probable that the friction due to the parts in question on the multi-cylinder models forms a much smaller proportion of the total loss and will furnish a correspondingly less productive field for the reduction of frictional losses.

The power required to operate the valves was so small as to be immeasurable under the conditions of the test and with the apparatus used. It is probable, however, that considerably more power is required under actual operating conditions, as the exhaust valve must be opened against high pressure. Some form of balanced valve would be necessary in order to reduce this loss.

In applying the results of these tests to the multi-cylinder Liberty engines, it should be borne in mind that the total mechanical loss will be considerably less per cylinder than on the single-cylinder engine. This is undoubtedly due to the proportional decreases in the crankshaft, camshaft and auxiliary friction as explained in a previous paragraph. However, the piston and ring friction and pumping losses probably remain about the same, irrespective of the number of cylinders. The curve (Fig. 9) is a comparison of the mechanical losses of the single-cylinder Liberty engine with those of the 6-, 8- and 12-cylinder types. These curves are not strictly comparative, due to the somewhat different conditions under which the tests of the multi-cylinder engines were run, but the curves are included to give an idea of the general tendency of the mechanical losses in these engines.

Description of Single-Cylinder Engine

The engine used in this test was designed to give approximately the same results as would be obtained from one cylinder of the Liberty aviation engine. It has a standard Liberty cylinder mounted vertically and fitted with standard valves, valve springs, cams, pistons and piston rings. The connecting-rod, crankshaft, crankcase and valve gear were designed to provide a serviceable engine for experimental work in connection with this cylinder. The accessory drive shafts and such auxiliary units as the water pump, oil pump, generator and carbureter are of similar design to those used on the multi-cylinder engines, but are of lower capacity. In spite of these changes, however, the actual conditions occurring within the cylinder itself will be substantially the same as in one cylinder of the multi-cylinder Liberty engines. Since it is a single unit, conditions occurring therein may be more closely studied. The accompanying photographs (Figs. 1 and 2) show the location of the various engine parts.

In assembling the engine, a cylinder that was in first-class condition and well worn-in was used, as it was decided that all piston ring tests are to be run with this kind of a cylinder. It is also planned to use a new piston and new rings in all piston ring tests, and to this end a new piston was fitted with a set of standard Liberty engine rings.

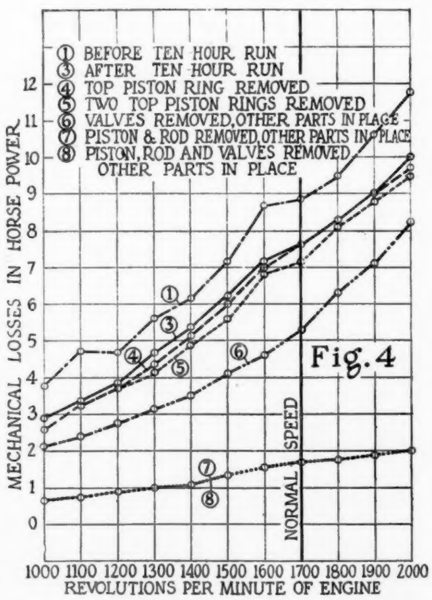


Fig. 4—Single-cylinder Liberty No. 6 curves, showing mechanical loss due to various parts

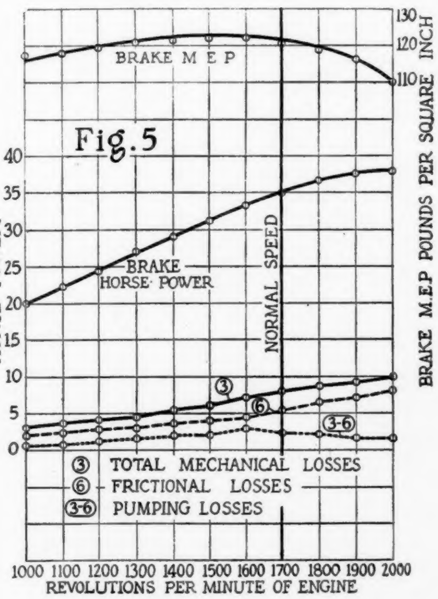


Fig. 5—Brake M.E.P., brake horsepower, total mechanical losses, frictional losses and pumping losses at various engine speeds

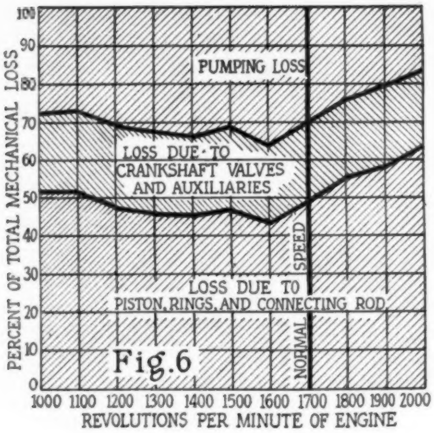


Fig. 6—Diagram showing frictional and pumping losses as part of the total mechanical loss

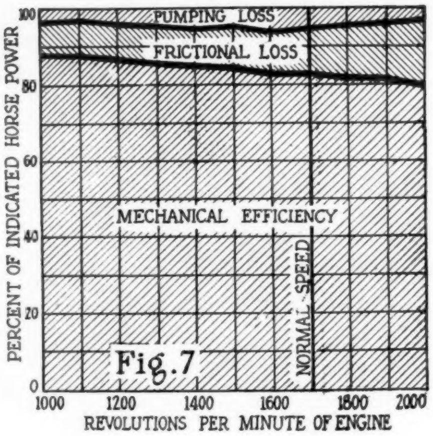


Fig. 7 — Diagram showing mechanical efficiency and losses at various engine speeds in per cent of indicated horsepower

Rings were selected which would bear in the cylinder over their entire circumference. The rings were then fitted in the piston with the following clearances:

Clearance of top ring in groove, 0.003 in.

Clearance of second and third ring in groove, 0.002 in.

Gap of all rings when in place in cylinder, 0.030 in.

The drawing (Fig. 3) shows the location of the above clearance, together with the clearances of the piston in the cylinder on all standard Liberty engines. The engine was then assembled and connected to an electric cradle dynamometer in the usual manner. The following runs were made:

1. Friction horsepower immediately after setting up.
2. One hour "running-in" by the dynamometer and one hour under engine's own power, gradually increasing the throttle opening until full power was reached. This was followed by a 10-hour full-power run at normal speed.
3. Friction horsepower after 10-hour run.

The above program has been laid out as a standard method of testing piston rings. The method of conducting the friction horsepower runs is to warm the engine up by circulating hot water through the jackets until the jacket temperature reaches the point at which it is recommended to run the engine. In case of the Liberty engine this is 170 deg. Fahr. When an engine is running under its own power, the oil temperature is such a variable that it is considered that the most comparative results will be obtained by allowing the oil to remain at room temperature. The oil, therefore, is not warmed up but is allowed to remain at room temperature. This was approximately 60 deg. Fahr. during the tests reported herewith.

When the jacket temperature had reached the required point, the engine was motored over by means of the dynamometer, at speeds varying from 1000 to 2000 r.p.m. as shown by the attached data sheets. The ignition current and gasoline were shut off, and the throttle set in the wide-open position, as is customary in friction horsepower runs. Readings of the speed, brake load, compression pressure, water inlet and outlet temperatures, oil pressure and barometer were taken at each speed. After the first friction horsepower run, the engine was motored over at 1000 r.p.m. for 1 hour by means of the dynamometer. It was then started under its own power, idling at first, after which the throttle was gradually opened until at the end of 1 hour the engine was operating at full power and normal speed. The 10-hour full-power run was made according to the standard method used by this laboratory. The engine was run at 1700 r.p.m. with wide-open throttle and readings of the speed, brake load, water temperatures and oil pressure, were taken at intervals of 15 min. throughout the run. The outlet water temperature was held at 170 deg. Fahr. and the barometer pressure was observed hourly.

The curves (Fig. 4) show the results of the above tests. It will be noted that the difference in friction horsepower before and after the 10-hour run is considerable.

Test to Determine Mechanical Losses

The general plan for the determination of the mechanical losses in various engine parts was to motor the engine over electrically, as in a friction horsepower run, with different parts removed. The difference in power required with and without the part in question was considered to be the amount of power absorbed by that part. The tests already made to furnish a basis for piston ring comparison tests also form the first part of this test. The "running-in" and 10-hour runs were considered sufficient so that the engine was worn in to represent service conditions. The friction horsepower run immediately after the 10-hour run furnished the basis of comparison with the other runs in which the various parts were removed. In addition to these runs, the following runs were made:

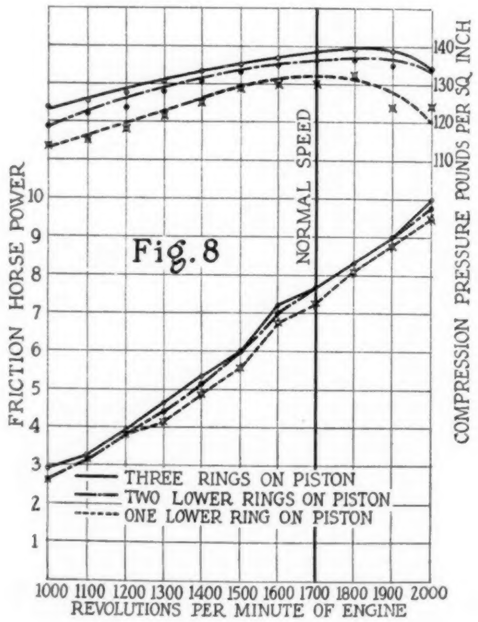


Fig. 8 — Friction horsepower and compression pressures with the piston fitted with one, two and three rings

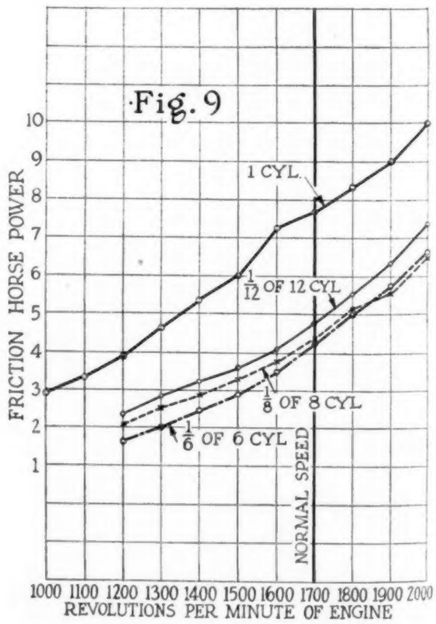


Fig. 9—Comparison of mechanical losses in Liberty single-cylinder with Liberty multi-cylinder engines

No. 4. Friction horsepower, top piston ring removed.

No. 5. Friction horsepower, 2 top piston rings removed.

No. 6. Friction horsepower, valves removed, all other parts in place.

No. 7. Friction horsepower, piston and rod removed, all other parts in place.

No. 8. Friction horsepower, valves, piston and rod removed, all other parts in place.

No. 9 Friction horsepower, engine reassembled.

No. 10. Power curve at full throttle.

The method used in making tests 4 and 9 were identical with that already described for making friction horsepower runs, except, of course, that compression pressure could not be measured in runs 6, 7 and 8. Run No. 10 was made by the standard method for taking full power curves.

The results of this test are given by the curves in Figs. 4 to 9. With a view toward enabling the reader to more clearly understand these, an explanation of the terms used will be necessary.

The "Friction Horsepower," so called, of a complete engine is assumed to represent the total mechanical loss of that engine when running under its own power. This mechanical loss is made up of two kinds of losses, viz., "frictional losses," which are due to the frictional resistance of the moving parts, and "pumping losses," which are caused by the work done in drawing in and exhausting the charge and by the heat lost to the cylinder walls.

The method of determining the indicated horsepower at any given speed was to add the friction horsepower to the brake horsepower at that speed.

To determine the total frictional losses, the valves were removed and the engine motored over with all other parts in place (run No. 6). As the power absorbed by the valves themselves was found to be negligible, this furnished a very close approximation of the actual frictional losses.

The pumping loss was determined by subtracting the total frictional loss from the total mechanical loss as found in run No. 3. The pumping loss thus found includes both the loss due to drawing in and exhausting the charge and the "thermodynamic" loss which is caused by that part of the heat of compression which is given up to the cylinder walls and not returned to the charge. These two parts of the pumping loss were not determined separately, and are, therefore, given together under the heading "Pumping Losses."

The frictional losses due to the various parts were determined, as already stated, by removing the part in question, and observing the friction horsepower with the part removed. The frictional loss due to any part is determined by the difference in power required to motor the engine over with and without that part. The results given herewith should not be considered as representing exactly the losses which occur in an engine when it is operating under its own power. Due to the lower viscosity of the oil, the generally higher temperatures throughout the engine and the higher bearing pressures on the principal moving parts, the frictional losses will be somewhat different when the engine is running under its own power and when it is being motored over by the dynamometer, in spite of the fact that the water jacket temperature is the same. The pumping losses will also differ between these two methods of operation on account of the difference in temperature of the gases and the difference in heat lost to the cylinder walls. It is considered, however, that the results contained in this report are reliable from a comparative viewpoint, and in this respect they represent a close approximation of actual working conditions.

Referring to the curves (Fig. 5) it is seen that the frictional loss curve increases at a greater rate than the engine speed. This is to be expected, as the frictional forces are dependent primarily on the forces acting on the piston and connecting-rod. These forces increase with the engine speed and the power loss, which is proportional to force multiplied by speed, will therefore increase at a greater rate than the speed.

From the curves it is also seen that the pumping losses decrease above 1600 r.p.m., which is probably due to decreased volumetric efficiency. This is further indicated by the fact that the brake mean effective pressure reaches its maximum at about the same speed as the pumping loss.

Evolution of High-Speed Steel

THE evolution of "high-speed steel" from "plain" water-hardening steel is undoubtedly due to Robert Forrester Mushet, his patent of 1858 becoming a practical manufacturer's realization in Sheffield about 1870, writes J. D. Arnold in an article in *The Engineer.* His first crude alloy contained nearly 2 per cent each of carbon and manganese, up to 9 per cent of tungsten, a moderate amount of chromium, and some tenths per cent of silicon and low sulphur and phosphorus. The first heat treatment applied was the crude method of spontaneously hardening in air from a red heat. This, however, was replaced by Mushet in the late eighties by hardening in air from a pale yellow heat, although this method was kept more or less secret. The gradual development of modern high-speed steel was marked by the general lowering of the carbon by about two-thirds—to, say, 0.6 per cent—and the manganese to 0.25. The tungsten was raised to 20 per cent and the chromium to 4 per cent. Silicon up to 0.5 per cent and sulphur to 0.1 per cent was also employed, this quantity not injuring the steel in the presence of low phosphorus.

The Bethlehem Company, of America, in 1900, showed at the Paris Exposition high-speed steel turning soft steel at a just visible red heat, this fact constituting a remarkable advance, which must be placed to the memory and metallurgical credit of Messrs. Taylor and White. About 1892-3 Sheffield high-speed steel with about 0.6 per cent carbon, 4 per cent chromium, and 20 per cent of tungsten marked the high-water mark in the quality of high-speed steel. The power of this steel was, however, doubled about 1896-7 by adding to it about 1 to 1.5 per cent of vanadium. The new C-Cr-W-Va alloy was capable of running dry on a hard shaft for several minutes at a fair red heat.

The patent taken out by the present writer in 1918 substituted, say, 6 per cent of molybdenum *associated with a little over* 1 *per cent of vanadium, the key element of the chemical composition.* With this composition and the proportion of other elements specified in the patent a steel distinctly better than the 1896-7 steel was obtained, always proving that the water-hardening heat treatment described in the patent is also carefully carried out. Hence, less than a third of the percentage of molybdenum compared with that of tungsten is required. It is therefore obvious that if the price of molybdenum can be reduced to that of tungsten a very considerable saving of cost and a complete abolition of the use of tungsten can be brought about. The functions of the key element vanadium appear to be: (*a*) stabilization of the variable properties of molybdenum steel, and (*b*) prevention of cracking during the water hardening operation, thus securing a steel, not only of notable hardness, but also of *remarkable thermal stability.*

Lubrication Control in Conformity with Engine Load

A by-pass valve actuated by the intake manifold vacuum, used as a means of lubrication control in a full-pressure oiling system, is discussed in this article. This type has proved satisfactory in service and may become general with full pressure lubrication.

By Fred C. Ziesenheim*

IN an internal combustion engine lubricated by a continuous force feed or pressure system, the oil is supplied to the bearings under pressure by a gear or displacement type of oil pump. A safety pressure release will be fitted which will limit the maximum pressures that can be developed and will prevent injury to the oiling system resulting from excess pressure.

Starting an engine in winter when the oil will be very viscous, and possibly at a lower temperature than its pour or cold test, may cause a rupture in the oil line, leakage at the unions, or other failure, resulting from the abnormal pressure.

If the engine is idling or carrying a light load with the throttle nearly closed, during the suction stroke, the vacuum within the cylinder will be at its maximum value, and if a generous supply of oil is thrown off the crankshaft onto the cylinder walls, there will be a tendency for the oil to be drawn into the combustion chamber, causing the formation of carbon, fouling of spark plugs, sticking of piston rings, and other attendant evils.

If no control device is provided, the quantity of oil delivered by a gear or displacement pump will be approximately proportional to the engine speed. (See Fig. 4.) On engines fitted with a centrifugal governor for speed control, the idling speed will be greater than the full load or rated speed, depending on the percentage regulation, hence more oil will be delivered at no load than at full load, whereas for correct lubrication the quantity supplied should be proportional to the load on the engine.

Control of the lubricating oil pressure is necessary if the quantity supplied is to be proportional to the engine load and if over-lubrication is to be prevented at no load or idling speeds.

An investigation of engine characteristics in a search for an engine function or force that will vary in accordance with the engine load and that can be utilized for maintaining the oil pressure in conformity with engine load discloses two variables—throttle opening, and intake manifold pressure.

The carbureter throttle opening varies with the engine load and has been used for controlling oil pressure. Examination of the throttle opening curve of Fig. 1 and the data of Fig. 3 shows that the throttle opening does not vary proportionately with engine load; therefore, throttle opening is unsuitable for lubrication control.

An additional objection to the operation of the lubrication control device by the throttle is that the device will be an additional resistance upon the governor and will produce an excessive speed variation.

The other variable, intake manifold pressure, expressed as a vacuum in millimeters of mercury, varies inversely as the engine load. Examination of the curves of Fig. 1

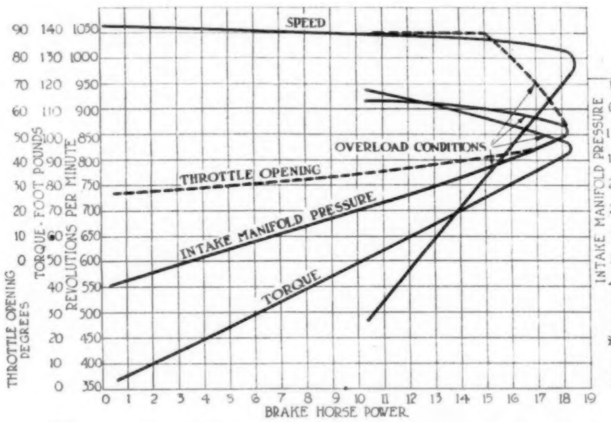


Fig. 1—Variation of throttle opening and intake manifold vacuum with engine load. Speed control by governor

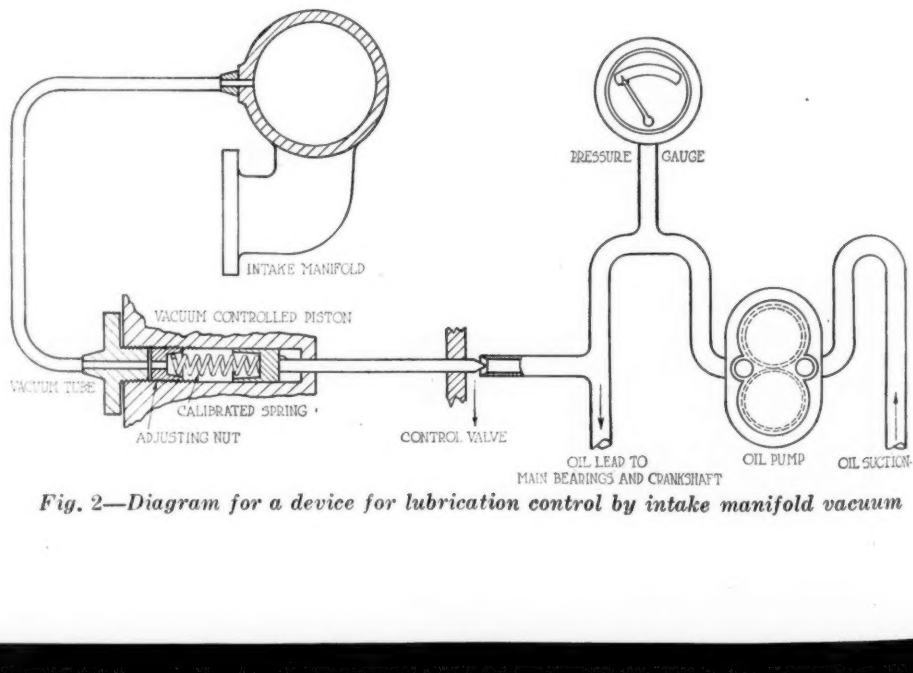


Fig. 2—Diagram for a device for lubrication control by intake manifold vacuum

**Mr. Ziesenheim is a member of the Society of Automotive Engineers and has contributed previously.*

Brake Horse Power	Per cent	0	25	50	75	100	....
	Value	0	4.63	9.25	13.88	18.5	11.5
Torque	Per cent	0	24.2	48.4	73.7	100	120
	Value	0	23	46	70	95	114
Intake Manifold Vacuum	Per cent	0	20	4.35	66.4	100	119
	Value	410	348	275	204	100	40
Throttle Opening	Per cent	0	11.1	25.9	44.4	100	237
	Value	26	29	33	38	53	90

Fig. 3—The percentage variation of intake manifold pressure and throttle opening with engine load, on a basis of the full load condition is 100 *per cent*

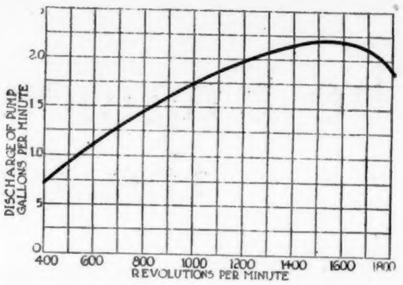


Fig. 4—Discharge capacity of a gear type pump for a full pressure oiling system

and the data of Fig. 3 indicates that as the engine load increases the vacuum existing in the intake manifold becomes less or approaches atmospheric pressure, at approximately the same rate. Therefore, the oil pressure, and likewise the quantity of oil delivered to the bearings, can be regulated to conform to engine load, by a suitable pressure release or by-pass in the oil line, controlled and operated by the intake manifold pressure.

The control device (See Fig. 2) may consist of a cylinder and piston, the chamber of the cylinder being in communication with the intake manifold through a tube connecting them. The piston rod extension will constitute or actuate a pressure relief valve composed of a conical valve and seat on a by-pass line tapping the oil delivery pipe just previous to the engine bearings. Movement of the piston due to the vacuum existing in the chamber will open the by-pass valve and allow the oil to pass freely into the crank case, thus relieving the oil pressure and limiting the quantity delivered to the bearings. The piston movement will be resisted by a calibrated spring which can be readily adjusted to give the desired range of oil pressures.

The control device described will have three functions to perform:

(a) It will act as a safety pressure release and prevent the occurrence of any abnormal oil pressures.

(b) It will prevent over-oiling from medium load to no load conditions.

(c) It will supply maximum lubrication at all speeds with the engine under full load.

The present practice in lubrication control by intake manifold vacuum is for the control device to cut out of operation at approximately three-quarter load, giving maximum lubrication to loads in excess of that. Lubrication control is imperative only at light and no loads when piston pumping and low intake pressures may cause an excessive quantity of oil to be drawn into the combustion chamber. The lubrication system must be designed so that maximum oil pressure will not produce over-lubrication at three-quarter load or more.

In service, the results obtained in controlling lubrication by intake manifold pressure have been satisfactory, and judging from the present design tendencies, the practice may become general with full pressure lubrication.

New High Magnesium Alloys

A SERIES of magnesium alloys, all containing more than 80 per cent by weight of magnesium, with slight additions of other metals such as zinc, is produced by the Chemical Works Griesheim-Elektron in Frankfort-on-Main, Germany. The alloy is silver-white and similar to aluminum in appearance. On account of its low specific gravity, it is particularly suited for castings or pressings for articles and machine parts which must combine high tensile strength with exceptionally low weight.

According to an article in *Stahl und Eisen,* after considerable difficulties had been overcome, it became possible to pour the metal, and any part which can be sand-cast of aluminum can also be cast of Elektron. The alloy, which is furnished in blocks of 4.4 weight, is melted down in crucibles of wrought iron or cast steel, of from 350 to 700 cu. in. capacity. It should be poured as soon as it has been completely melted. Only in the case of thin-walled or very large castings, is it necessary to slightly overheat the metal. The melting temperature is in the neighborhood of 1165 deg. Fahr. and the alloy should never be heated more than 100 deg. Fahr. beyond this, as otherwise the layer of oxide floating on the surface will catch fire. In order to extinguish a fire that has once started, there is added to 100 parts of Elektron alloy from two to three parts of a calcium alloy. The calcium has a protective action and prevents the oxidation for a time; on the other hand, it is objectionable because it makes the castings brittle and subject to atmospheric influences.

In view of the sensitiveness to overheating of the alloy, it is advantageous to pour thin-walled and large size castings in molds which are comparatively hot, that is, which come directly out of the drying oven. Elektron castings show a mean tensile strength of 17,000 to 21,000 lb. per sq. in., and an elongation of 3 to 4 per cent. In sand mold castings, which cool more slowly, the tensile strength is from 3000 to 4000 lb. per sq. in. less, and the elongation about 3 per cent. The reduction of area has about the same value as the elongation. The limit of proportionality lies between 5700 and 7000 lb. per sq. in., the elastic limit between 12,000 and 14,000 lb. per sq. in. Hardness tests with the Shore Scleroscope gives values of 10 to 15.

Slip-Stream Corrections in Performance Computation

REPORT No. 71 of the National Advisory Committee for Aeronautics deals with the variation of slip-stream velocity with the rate of advance of a propeller, and treats the subject both from the experimental and the analytical standpoint. The experimental portion is based on Eiffel's work. The relative increase in slip-stream velocity is much less rapid than that in air-speed, and the slip-stream correction therefore falls off as the air-speed increases. A method, based on the momentum theory of propulsion, is given for determining the slip-stream correction for any given propeller under any conditions. The curve of slip-stream velocities thus determined by theory checks extremely well (within 2 per cent) with the experimental curve for the one propeller for which data are available.

The Engineering Fundamentals of the Crawler Tractor

This article by Mr. Jandasek was written for the purpose of furnishing a basis for working out the necessary mathematical problems that arise in the design of this type of tractor. It is a field possessing but little technical literature and this article is a thorough exposition of its subject.

By Joseph Jandasek, M. E., E. E.*

WHEN a crawler rolls over soft ground, it is supported by a surface of $F\ H'$ sq. in. area, where F is the face of the crawler and H' the wheelbase, both in inches. Assuming that the reaction of the soil against the crawler increases in direct proportion to the depth of depression d, this reaction being nil at the surface and q lb. p. sq. in. when the soil is depressed 1 inch, we have for the total carrying capacity of the crawler

$$W = q\ d\ F\ H' \text{ lb.} \qquad (1)$$

When we know the depression we can calculate the carrying coefficient of the soil q in lb. per cu. in. and we may then write

$$d = \frac{W}{q\ F\ H'}$$

On the basis of the above assumption, we come to the conclusion that the work which must be performed to cause the depression d is

$$K = \frac{W}{2}\, d = \tfrac{1}{2}\, q\ d^2\ F\ H' = \frac{W^2}{2q\ F\ H'} \text{ in.-lb.}$$

This amount of work must be done in each distance H'; therefore,

$$tW\ H' = \frac{W^2}{2q\ F\ H'}$$

where tW is the pull required to propel the vehicle over level ground; in other words, its rolling resistance. Hence, the coefficient of rolling resistance for crawlers is

$$t = \frac{W^2}{2q\,F\,H'^2} \text{ lb. per lb} \qquad (2)$$

The resistance increases in direct proportion to the weight per inch of width and inversely as the square of the wheelbase. From this it can readily be seen that long tracks are very advantageous. The carrying coefficient is not the same as the corresponding value q for wheel tractors, as used by the writer, for instance, in equation (52) in an article which appeared in AUTOMOTIVE INDUSTRIES of June 12, 1919, but must be determined separately for each particular type, as well as for each size of tractor. Nevertheless, the formula is useful in giving the designer an idea regarding the relations between the crawler dimensions and the rolling resistance. The area of ground contact of wheels as well as crawlers ought to be the largest possible in order to reduce the rolling resistance and to insure against miring.

Engineer for the Paige-Detroit Motor Car Co.

Influence of crawler dimensions on the available drawbar pull. In order to obtain the greatest possible amount of ground adhesion, the common practice is to equip the crawler with driving spurs; the resistance to slippage is then due to friction and cohesion of the earth strata and not to friction of the crawler surface on the earth. In order to drive the spur (Fig. 1) into the soil and at the same time to pull the load, a definite amount of pulling weight w_1 and dead weight w_2 is necessary. After the spur is in the ground, pulling weight w_1 for each spur is necessary in order to hold it down and counteract the tendency of the drawbar pull to pull it out of the ground. Then the total weight

$$W = w_2 + n\ w_1$$

and the total available pull

$$P = n\ f\ w_1 + c\ F\ H'$$

where:

n = number of spurs,
f = coefficient of friction of earth on earth,
c = coefficient of cohesion.

Evidently the weight w_2 is not producing any pull; therefore, leaving the supporting area of crawler ($F\,H'$) the same, we can increase the available pull P by making the tracks narrower and longer, with the result that the weight w_2 necessary to drive the spurs into the soil is smaller. Thus long and narrow crawlers possess better weight efficiency than wide ones.

Crawler design. As the front wheel drive tractor has a poor weight efficiency (the more it pulls the less weight is supported by the driving wheels) crawler tractors are generally made rear wheel driven. In Figs. 2 to 6 are illustrated three different ways of arranging the tracklayer:

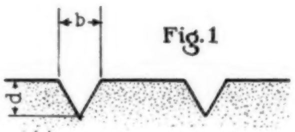


Fig. 1

1. Tractors with two crawlers in the rear and steering wheels in front,

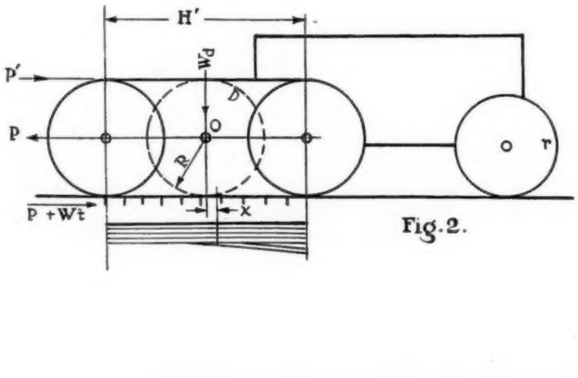
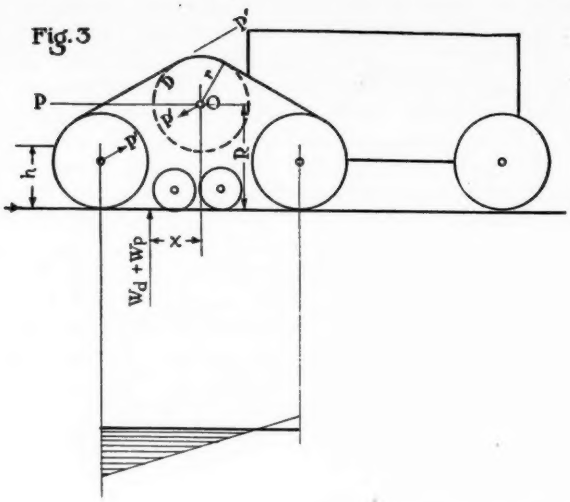


Fig. 2.

Fig. 3



the machine being suspended in the middle of the tracks so that the tracks can oscillate around the rear axle *O* (Figs. 2 and 3);

2. Tractors with two crawlers and separate steering wheels, with the point of suspension at one end of the track (Figs. 4 and 5);

3. Tractors with merely two crawlers, without any steering wheels (Fig. 5).

The first problem the tractor designer is confronted with is that of weight distribution and proper balance of the crawlers alone. Let us investigate now the equilibrium of external forces acting on the crawler shown in Fig. 2. The driving force P' acting at the point where the track chain is being pulled by the teeth of the driving sprocket, the wheel *D* has a tendency to swing the whole track around the rear axle clockwise with a moment $+ P'R$ in.-lb. The reactive force of the soil on the spurs has an anti-clockwise moment $- PR$. The reaction necessary to drive the track over the ground and acting on the periphery of the driving sprocket *D* has a moment $- Wx$; the force propelling the front wheels over the ground is $- W_f tR$; the friction of the chains and wheels is $- fR$. Further

$$P' = P + Wt + f$$

For equilibrium of the crawler alone we have

$$P'R = PR + W_d x + fR + W_f tR,$$

and finally for the distance x from the center of the track to the resultant of the ground reaction we obtain

$$x = Rt \qquad \text{.........(3)}$$

Thus we come to the conclusion that the resultant of vertical reactive forces of the earth is not going through the rear axle *O*, but is moved a distance Rt ahead. (The weight of a tracklayer in motion is not supported directly under the rear axle, but a distance Rt ahead of it—the same as a wheel type tractor when in motion.) The pressure under the crawler in Fig. 2 is not equally distributed, being slightly higher at the front and decreasing gradually to the rear. The specific pressures can be found by the following equations:

$$w_{max} = \frac{W_d + W_p}{F\ H'}\left(1 + \frac{6\,Rt}{H'}\right) \text{ lb. per sq. in.} \qquad \text{........ (4)}$$

$$w_{min} = \frac{W_d + W_p}{F\ H'}\left(1 - \frac{6\,Rt}{H'}\right) \text{ lb. per sq. in} \qquad \text{.....(5)}$$

where H' represents the wheelbase of the track.

In the case represented in Fig. 3 the positive moment is $P'r$, while the negative moments are

$$PR + W_d x + fR + W_f\,tR$$
$$P' = P + Wt + f$$

and from equation of the equilibrium we obtain:

$$x = Rt - \frac{(P + Wt + f)\ (R - r)}{W_d} \qquad \text{.........(6)}$$

The specific pressures may be found in the same way as in the above case, substituting the value of x from equation (6) in equations (4) and (5). It is very important to check these specific pressures, otherwise the design developed might turn out very inefficient.

The track shown in Fig. 4 is free to oscillate around the rear axle *O;* consequently there would be no pressure except under the rear axle. Nevertheless, as soon as the tractor moves forward, a slight pressure under the whole length of the crawler would develop, because the center of support moves ahead a distance Rt. This specific pressure, however, would not be sufficient, and, therefore, the spring *S* is applied in order to add more weight to the front of the track. The case illustrated in Fig. 5 is not correct from the standpoint of pressure distribution; this tractor would have a better pressure distribution if its direction were reversed.

In the third case (Fig. 6) there is the tendency already discussed of shifting the weight from the front of the tractor to the rear, because of the drawbar pull reaction. The pressure under the crawler, too, is unevenly distributed, being again a minimum in front and a maximum in the rear. Applying the law of equilibrium of external forces for the case when the tractor is pulling on the level we obtain:

$$Ph - Wa + xW = O$$

$$x = a - \frac{P}{W}h \qquad \text{.........(7)}$$

x being measured from the rear axle; and for specific soil pressures we have:

$$w_{max} = \frac{2\,W}{H'\,F}\left(2 - \frac{3x}{H'}\right) \text{ lb. per sq. in.} \qquad \text{.....(8)}$$

$$w_{min} = \frac{2\,W}{H'\,F}\left(\frac{3x}{H'} - 1\right) \text{ lb. per sq in.} \qquad \text{....(9)}$$

Examples:

Find the proper pressure distribution under two crawlers pulling a maximum load on the level for the following tractor (Fig. 2): $W = 3000$ lb., $H' = 50$ in., $F = 6$ in., $H = 80$ in., $P_o = W$, $R = 15$ in., $c = 22$ in., $h = 15$ in., $t = 0.20$, $g = 0.10$, $A = 30$ deg.

Fig. 4

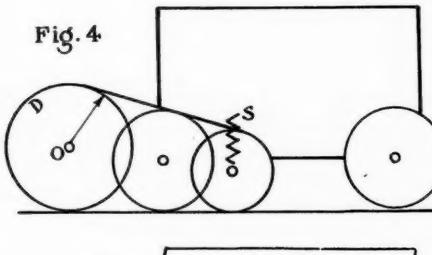
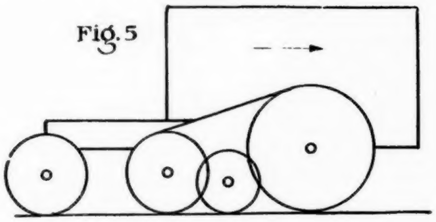


Fig. 5



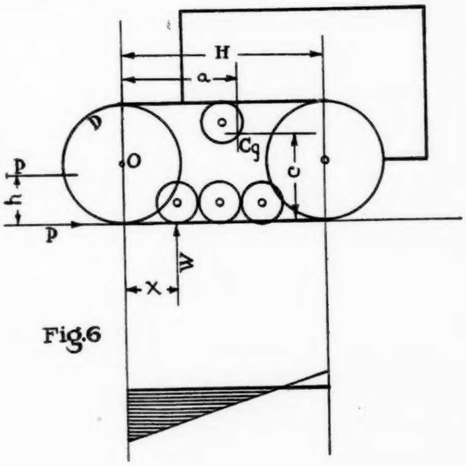


Fig. 6

Distance a for center of gravity:

$$a \times 0.866 = (1 + 0.10 - 0.5) \times 15 + 24 \times 0.5 + 15 \times 0.20 \times 0.87$$

$a = 27.2$ in., or in round figures 28 in.

Weight distribution at standstill on level:

$$W_d = 3000 \times \frac{52}{80} = 1950 \text{ lb. on rear}$$

$$W_f = 3000 \times \frac{28}{80} = 1050 \text{ lb. on front wheels}$$

Weight on rear when pulling full load on level:

$$(W_d + W_p)\ (H - Rt) = W\ (H - a) + P_o h$$

$$W_d + W_p = \frac{3000 \times 52 + 3000 \times 15}{80 - 3} = 2600 \text{ lb.}$$

$$W_f - W_p = \qquad 400 \text{ lb.}$$

Average specific pressure =

$$\frac{2600}{6 \times 50 \times 2} = 4.35 \text{ lb. per sq in.}$$

Maximum specific pressure =

$$4.35 \times \left(1 + \frac{6 \times 3}{50}\right) = 5.9 \text{ lb. per sq. in.}$$

Minimum specific pressure =

$$4.35 \times \left(1 - \frac{6 \times 3}{50}\right) = 2.8 \text{ lb. per sq. in.}$$

Now let us consider another example (Fig. 3) in which all data are the same as in example 1, except that $r = 10$ in., $R = 24$ in., $f = 200$ lb.

According to equation (6)

$$x = 4.8 - \frac{(3000 + 600 + 200) \times 14}{3000} = -12.9 \text{ in.}$$

Average specific pressure = 4.35 lb. per sq. in.

Minimum specific pressure =

$$4.35 \left(1 - \frac{6 \times 12.9}{50}\right) = -\ 3.2 \text{ lb. per sq. in.}$$

Maximum specific pressure =

$$4.35 \left(1 + \frac{6 \times 12.9}{50}\right) = +\ 11.9 \text{ lb. per sq. in.}$$

Example 3 (Fig. 6):

Let $W = 3000$ lb., $P_o = 3000$ lb., $H = 50$ in., $F = 6$ in., $R = 15$ in., $c = 22$ in., $h = 15$ in., $t = 0.20$, $g = 0.10$, $a = 28$ in.

According to equation (7)

$$x = 28 - 15 = 13 \text{ in.}$$

Average specific pressure = 5 lb. per sq. in.

Maximum specific pressure =

$$2 \times 5 \left(2 - \frac{3 \times 13}{50}\right) = +\ 12.2 \text{ lb. per sq. in.}$$

Minimum specific pressure =

$$2 \times 5 \left(\frac{3 \times 13}{50} - 1\right) = -2.2 \text{ lb. per sq. in.}$$

From the standpoint of low soil pressures and favorable distribution of the same example 1 shows the best result. In example 2 the radius r must not differ too much from the height R; otherwise high specific soil pressure results and the stability of the track may even be endangered.

Wheelbase calculation of two-crawler tractors. Let us consider now the case of a machine descending a steep grade A, when it is liable to tip over forward. From Fig. 7 it will be seen that in a two-crawler tractor the conditions are most favorable to tipping over forward when the machine is descending a steep hill and not pulling any load. If it were not for the radius of the front crawler idler, tipping would commence when $H\text{-}a = c \tan A$. Here a is the horizontal distance of the center of gravity from the rear wheel axis, and the minimum value of a is limited by the necessity of preventing rearing at heavy drawbar loads. The height c of the center of gravity is determined by considerations of clearance required, and depends upon the construction of the machine; angle A should be taken equal to the maximum natural slope, in order to keep on the safe side. Therefore angle $A = 45$ deg. The only factor which can and must be varied to satisfy the condition of safety is wheelbase H

$$H = c \tan A + a \ldots\ldots\ldots\ldots (10)$$

Let it be required to design a crawler tractor of the following specifications: The machine has to pull two 14-in. plows, ascend gradients up to 10 per cent in soil with a ground resistance $f = 15$ per cent, equipped with 2 crawlers, without steering wheels, with 3 speeds forward and one reverse, the low speed S_l being 2 m.p.h., the plowing speed S_p, 3 m.p.h., and the high speed S_h, 8 m.p.h. In calculating the engine size we will use the N. A. C. C. formula, because it is simple and well known. This formula is based on a brake mean effective pressure of 67.2 per sq. in. We will call this the "normal" brake mean effective pressure. However, in order to have sufficient reserve power for tough places, the engine must be able to develop at least 80 lb. per sq. in., which we can call "maximum" brake mean effective pressure.

Solution—Since the normal draft of one 14-in. plow is about 750 lb., the drawbar pull at plowing speed must be

$$\text{Normal drawbar pull} = P_p = 1500 \text{ lb.}$$

$$\text{Maximum drawbar pull} = 1500 \times \frac{80}{67.2} = 1800 \text{ lb.}$$

The weight of crawler tractors is always larger than that of wheel type machines, because of their necessarily heavy track construction; we will assume the total weight W to be equal to twice the normal plowing drawbar pull. Hence the weight of the machine will be 3000 lb. Further, the normal pull at low speed, P_l, is 2250 lb., the maximum pull at low speed, $P_{l\ max}$, 2680 lb., and the total rolling resistance,

$$(t + g)\ W = (0.10 + 0.15)\ 3000 = 750 \text{ lb.}$$

The normal drawbar horsepower is

$$DHP = \frac{P_p}{375}\ S_p = \frac{1500}{375} \times 3 = 12 \text{ hp.}$$

We are not considering slippage here, as that is purely a loss of speed and does not affect the pull.

Assuming the transmission efficiency to be 85 per cent and sprocket wheels and chains to account for another loss of 10 per cent, we have for the efficiency of transmission from engine to wheels, $E = 75$ per cent. Then the normal brake horsepower is

$$BHP = \frac{P_p + (t + g)\ W}{375\,E}\ S_p = \frac{1500 + 750}{375 \times 0.75} \times 3 = 24 \text{ hp.}$$

Selecting a four-cylinder, four-cycle gasoline engine and assuming a piston speed of 900 ft. per min., which is com-

mon in tractor engines, we obtain for the normal brake horsepower:

$$BHP = 0.9 \frac{B^2}{2.5} \times 4 \quad \ldots\ldots\ldots\ldots (11)$$

from which we derive the bore

$$B = 0.833 \sqrt{BHP} = 0.833 \times \sqrt{24} = 4.08 \text{ in.}$$

and the stroke,

$$L = 1.3\,B = 5.3 \text{ in.}$$

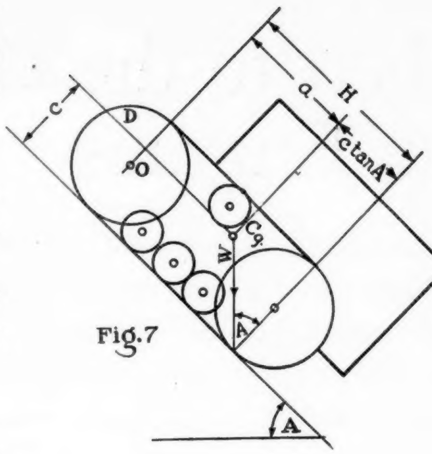


Fig. 7

We will make the bore 4 in. and the stroke 5½ in.

The speed of the engine,

$$N = 6 \frac{S_{piston}}{L} = 982, \text{ say } 1000 \text{ r.p.m.}$$

The piston displacement:

$$C = \pi B^2 L = 276 \text{ cu. in. (4-cylinder engine)}$$

There being one pair of miter gears between engine and pulley, the

$$\text{Belt } HP = 0.95 \times 24 = 22.8 \text{ hp.}$$

This allows of the use of an 18-in. grain separator.

If we make n_p (r.p.m. of pulley) equal to N (r.p.m. of engine) we get for the diameter of the pulley

$$d = \frac{10{,}000}{n_p} = 10 \text{ in.}$$

The face of the pulley can be made 6½ inches.

Selecting a 27-in. pitch diameter for the driving sprocket wheel (which gives an outside diameter of about 30 in.) we have for n, the r.p.m. of drivers,

$$n = \frac{S}{D} 336 \text{ r.p.m.}$$

and for G, the gear ratio:

$$G = \frac{N}{n}$$

Speed	Sprocket wheel r.p.m.	Gear ratio
2 m.p.h.	25	40
3 m.p.h.	37.3	26.8
8 m.p.h.	100	10

If $W = 3000$ lb., $P_o = 2680$ lb., $R = 15$ in., $c = 22$ in., $h = 15$ in., $A = 30$ deg., $g = 0.10$ and $t = 0.15$, the distance of the center of gravity from the driver axis in a horizontal plane is

$$a \times 0.866 = \frac{2680}{3000} + 0.10 - 0.5)\ 15 + 22 \times 0.5 + 15 \times 0.15 = 24 \text{ in.}$$

Assuming $e = 7$ in., we obtain for the same distance

$$a = \frac{2680}{3000} \times 7 + \frac{2680}{3000} \times 15 + 22 \times 0.10 + 15 \times 0.15 = 24.1 \text{ in.}$$

Assuming the maximum angle of descent $A = 45$ deg., the shortest safe wheelbase will be:

$$H = c \tan A + a = 22 + 24 = 46 \text{ in.}$$

In order to gain additional safety, we will take $H = 50$ in.

Assuming the mean pressure of earth contact to be 5 lb. per sq. in., the total crawler area will be 600 sq. in. This gives for the width of each crawler surface, $F = 6$ in. The specific pressure of ground contact is

$$x = a - \frac{P}{W} h = 24 - 13.4 = 10.6 \text{ in.}$$

The maximum specific pressure is

$$2 \times 5 \left(2 - \frac{3 \times 10.6}{50}\right) = 13.64 \text{ lb. per sq. in.}$$

and the minimum specific pressure,

$$2 \times 5 \left(\frac{3 \times 10.6}{50} - 1\right) = -3.64 \text{ lb. per sq. in.}$$

Uniform Dimensions Would Reduce Truck Building Costs

(*Continued from page* 1325)

In the actual laying out of the control units, while most manufacturers have given considerable study to this point in order to make driving and handling as easy as possible, others have neglected a great many points of importance. The seat height and steering wheel location should be such that the driver sits in an easy, natural position, and has good, clear vision. He should not be compelled to have to lean or reach to too great a degree for his shifter or brake levers. It may be necessary for him at any time to make a quick shift of gears, or to use the hand brake. In either case these should be readily and instantly available without requiring groping or too great an amount of reaching. It is, of course, not expected that the driver of a truck will demand the same amount of ease as the driver of a passenger car. Nevertheless, he occupies the seat for many hours per day and, consequently, everything possible should be done to reduce the fatigue element. A man gives better results and a truck costs less to operate when fatigue and discomfort are reduced to a minimum. Large fleet owners have made careful studies of this situation and have found that a material saving results and efficiency increased when the driver is made as comfortable as possible.

From the dimensions given in the tabulations herewith, it would seem possible to formulate a set of recommended dimensions. Their adoption by manufacturers would make matters much simpler for body builders, and also add to the comfort of the driver.

A COMPANY has been formed in London for the purpose of manufacturing calcium carbide and of extracting motor spirit, tar oil, and other by-products from coal and shale at Ballengeich Collieries, Natal, South Africa.

Carburetion as Applied to the Design of Automobile Engines

This article relates an unconventional theory of liquid fuel distribution phenomena, shedding light upon a problem to which the engine designer has given little thought but has left almost entirely to the carburetion engineer. But with fuel becoming heavier and of lower grade, certain problems are put up to the engine builder which he must solve himself.

By E. S. MacPherson*

IT is still true to a large extent that the carburetion of a new engine is left almost entirely to the carbureter manufacturer. The engine designer provides a hole of a certain size in the engine, which leads to the various inlet valves. Then, when the new model is built, the carbureter engineer does what he can with it. This would be very well if the carbureter alone were responsible for carburetion, but that is unfortunately not the case—and is less and less so as our gasoline becomes heavier and heavier.

We should demand of the carbureter the following:

1. Proper proportioning of the gasoline and air mixture for all steady speeds and loads.
2. Provision for richer mixture during acceleration without sacrifice of economical steady running mixture.
3. The maximum atomization with least restriction.

The above requirement can be approached very closely by carbureters obtainable to-day.

The carbureter, however, is not responsible for what happens to the air-gasoline mixture after it has passed on into the intake manifold and into the engine cylinders.

The qualitative distribution among the various cylinders and the condition of the mixture which enters the cylinders depends on the inlet pipe, providing the carbureter has first done its work properly. Thus the engine designer's carburetion problem is principally one of proper manifold design. What is meant here by manifold is the whole system of pipes from the carbureter flange to the various inlet valve seats. The varying quality of gasoline has forced the engine designer to give this problem greater attention. Available data on this subject are rare, except for gas speed data, and this is unreliable except for duplication of designs.

*Member of Society of Automotive Engineers.

Correct manifold design cannot b[illegible]ulated or plotted on curve sheets.

Engines having six cylinders, or a multiple of six cylinders, are considered harder to "carburete" than those having four or a multiple of four cylinders, but this is only because there are fewer possible manifold designs for this type, and thus fewer chances of mistakes. The almost universal type of four-cylinder manifold is very nearly correct as to shape, and for purposes of illustration the harder problem, the six, is made the subject of this article.

Carburetion troubles are principally apparent in cold weather and consist of the following:

1. Hard starting.
2. Missing.
3. Fouled spark plugs.
4. Gasoline in crankcase with attendant bearing trouble and smoky exhaust.
5. Poor economy.

No. 1 is not entirely a carburetion trouble. The others are all traceable to poor carburetion and can be eliminated by design for carburetion. The manifold must be so designed as to give equal "liquid" distribution, and we must also provide heat where it will be the most effective. The use of hot water heat is out of date, even with temperature control, because (1) it is not sufficiently intense, and (2) it is not immediately available after starting. We therefore should use exhaust heat, which is available as soon as the engine starts.

To explain what we mean by liquid distribution, we will say that of the gasoline which enters the manifold of a cold engine as liquid, each cylinder should receive an equal amount. Most of the gasoline in a cold engine enters the manifold in this form, and some also when the engine is warm, up to gas speeds of 4000 ft. p. m.

The application of exhaust heat or "hot spotting" is

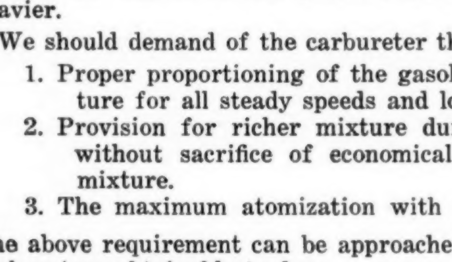


Fig.1

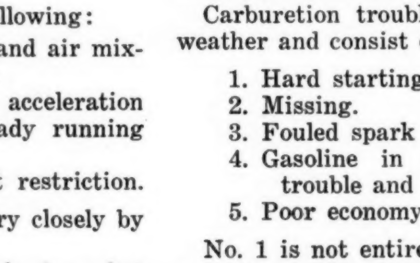


Fig.2

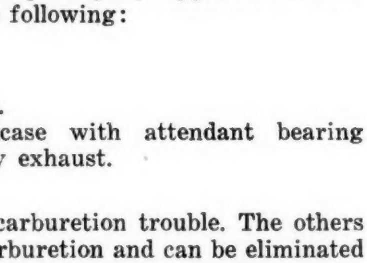


Fig.3

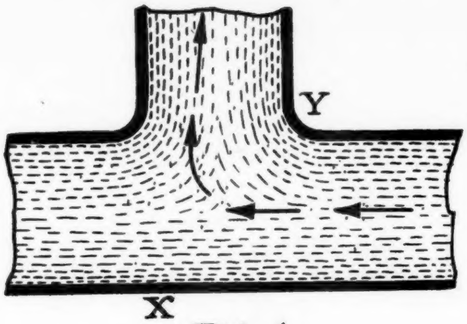


Fig. 4

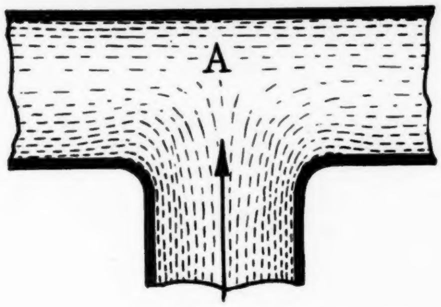


Fig. 5

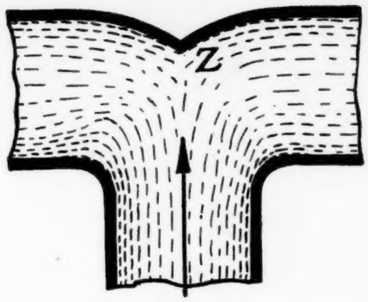


Fig. 6

coming to be recognized as a necessity, in spite of the objection frequently raised that exhaust heat expands the mixture, thus decreasing the power. This objection is not well taken, because when properly applied, exhaust heat will not decrease the maximum power, and will increase the power at lower speeds. Exhaust heat prevents "loading" and permits running on economical mixtures at all seasons of the year. In cold weather, owing to good vaporization, only a few minutes are required to make it possible for the engine to run on its normal mixture. Fig. 1 shows the change in characteristics accomplished in a $3\frac{1}{4} \times 4\frac{1}{2}$-in. engine, developed by the author, by hot spotting and liquid distribution. There was no other difference in the two engines. Curves *A* show the characteristics of the engine with a hot spot and redesigned manifold. Curves *B* show the characteristics of the engine before the change was made.

Liquid distribution as above mentioned affects the engine action most when cold, and even when warm affects it at lower speeds, that is, in hill climbing and acceleration, when the throttle is well open. It is, therefore, of importance that we have a properly shaped manifold.

To shape properly the manifold, the action of the entrained liquid gasoline in a cold engine should be understood.

First, we will consider the usual "T" beyond the carbureter as illustrated in Fig. 2. This is the only bend where the greater part of the liquid gasoline and the air impinge at the same point (*A*). There is, however, a small amount which trickles around the corners *B* and *C*. Beyond this "T" nearly all the liquid will run along the pipe walls fairly well distributed.

On passing a bend such as is illustrated in Fig. 3, nearly all the liquid will collect on the inside of the bend. This is contrary to the popular idea, but is, nevertheless, true. It is a fact that the deposition which takes place in the bend takes place at the outside, but the liquid gas coming to the bend seeks the inside, following the rule that heavy particles are slow moving and seek the region of least velocity. This segregation of gasoline and air will, of course, take place to an extent depending on the "sharpness" of the bend. Thus, the advantage of long, easy bends.

The branch riser shown in Fig. 4 is often used, and is the source of poor distribution. It is seen that the shape of the manifold at the branch is not such as to allow the entrained liquid to be carried around the bend and into the cylinder sucking from this branch. Some of this liquid is left at *Y* to be picked up wholly or in part by the next cylinder, drawing through the same header, thus enriching its charge. Branches as illustrated in Fig. 4 should be avoided. So should enlargements as shown in Fig. 5. The air speed decreases in the enlargement *A* and thus the carrying capacity of the air stream for unevaporated gasoline is decreased, causing deposition and a resultant "chunky" mixture, with poor economy and torque.

The type of "T" illustrated in Fig. 6 is successfully used in ventilating systems, but is not good practice in manifold design, for both sides of the "T" are not drawing at the same time, and even so the mixture is not sufficiently homogeneous to be divided in this manner.

The manifold illustrated in Fig. 7 has been extensively used, but is an extremely poor one. All the intermediate branches are of the unsuccessful type shown in Fig. 4, and it also has the "T" illustrated in Fig. 6. Cylinders 1 and 6 will get a mixture richer in liquid than 2 and 5, which in turn will get a richer mixture than 3 and 4. Also, on account of the type of "T" used, the engine will be very sensitive to small displacements of the carbureter on its flange. It might also be added that another reason for its failure is the unequal distances from the carbureter to the valves, making the conditions of deposition different for different cylinders.

Fig. 8 illustrates another common type of inlet pipe, and one which also has its faults. The three-way point at *C* constitutes oftentimes an enlargement such as shown in Fig. 5; also, 3 and 4 are nearer the carbureter than the other cylinders. Manifolds of this type usually require a high air velocity and cause lack of torque.

Fig. 9 shows a common type of manifold, where the carbureter is either attached at *A* or at a distance from *A*. In this manifold, the leads to cylinders 2 and 5 are of the undesirable type shown in Fig. 4. Also, the bends at *X* and *Y* are like the bend shown in Fig. 3, which fact causes cylinders 3 and 4 to get the liquid deposited at *X* and *Y* by the other cylinders. It is evident that if cylinders 1, 2 and 3 are to obtain equal amounts of liquid

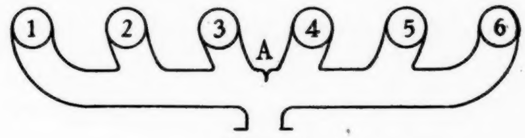


Fig. 7

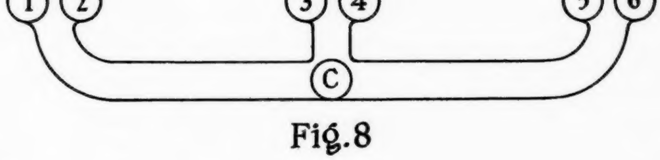


Fig. 8

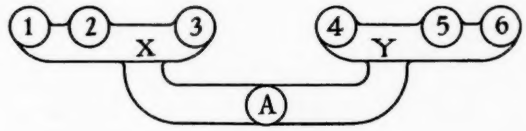


Fig. 9

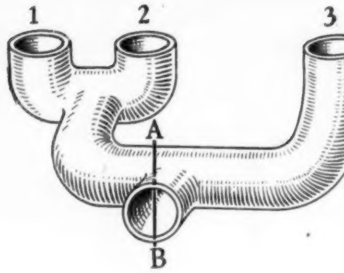


Fig. 10

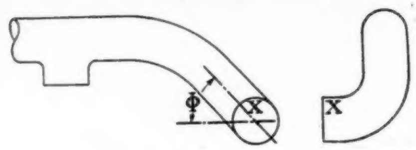


Fig. 11

gas, this gas must enter the "gallery" 1, 2, 3, or 4, 5, 6, symmetrically.

Fig. 10 shows the cored-in gallery for each group of cylinders, preferred by the author, and one which has been used with success. It will be noted that each valve is at a terminal and thus is in a position to draw the same amount of gas as any other of the group, provided this liquid enters the "gallery" symmetrically, that is, somewhere on the line *A-B*.

As to the pipe which connects the two gallery groups, which in this case will be exterior or bolted on, the author has found the "Ram's Horn" shape illustrated in Fig. 11 to be very satisfactory. The liquid gasoline will leave the end of this pipe on the vertical center line at *X*, Fig. 11, provided the "Ram's Horn" has been properly developed, and the two bends making this shape have the right relation. This is, of course, a matter of trial, but the angle φ will be about 45 deg.

With a six-cylinder manifold made up of two galleries, each feeding three cylinders, as shown in Fig. 10, and the outside manifold joining these two galleries, as illustrated in Fig. 11, we have the following advantages which tend toward good distribution, and which may be considered as rules therefor:

1. The distances along the inlet piping from the carbureter to the various cylinders are the same.
2. There are no sudden enlargements as shown in Fig. 5.
3. There are no branches of the type shown in Fig. 4.
4. No more than two branches lead from the same point.
5. The charge being drawn by any cylinder not only passes through the same length of pipe, but encounters the same number and kind of bends as the charge drawn by any other cylinder. Thus, the amount of gas lost by all cylinders by deposition, or that picked up by them, is the same. These conditions are those necessary for liquid distribution.

Assuming that the engine is now properly manifolded for good distribution, we still have to provide means for vaporizing sufficiently the heavy gasoline more or less finely atomized at the carbureter. As mentioned earlier, we must use exhaust heat, because of its intensity and immediate availability on starting the engine. The best place to apply this heat is at the carbureter "T" illustrated in Fig. 2. It must be applied to that portion of the pipe where liquid gasoline is present. If applied to a point such as the outside of the bend, shown in Fig. 3, very little of the gasoline will be heated and the heat applied will tend to expand the air impinging at this point. Also, if we apply the heat at the inside of the bend where a stream of liquid gas is found, there will be a tendency toward stratification of air and vapor, and also a very large quantity of heat will have to be applied, and even then there will be only mediocre results. Also, the application of heat here will not assist distribution at the "T" above the carbureter.

In short, what is necessary is that—

1. The heat should be applied as near as possible to the carbureter, where the heavy liquid is still atomized, and has not collected in streams on the walls of the inlet pipe.
2. It should be added before the gas passes any branch, so that the heat may assist the distribution for the whole motor.
3. The heat should be applied where it can be aided by the pulverizing action of the air on the gasoline.

In order to satisfy the above conditions, we must provide the hot point at the carbureter "T" illustrated in Fig. 2. The great portion of this heat must be provided on the upper portion of the "T," for it is here that most of the liquid coming from the carbureter strikes, as well as the air current. As mentioned earlier, there is, however, a small amount which trickles around the corners *B* and *C*, respectively, and which, if allowed to go unheated, will cause loading at the lower engine speeds. Heat must, therefore, also be applied at these two points. Fig. 14 illustrates the successful application of heat in this manner. This application has become known as the "hot spot." An opening in the exhaust manifold, preferably opposite one or more exhaust ports, is covered by the hot spot, which receives heat by radiation and from the impinging hot gases. The application of heat through a tube of any appreciable length is not sufficiently effective.

The gas speeds to be used naturally depend on the service to which the engine is to be put. Data from other engines are not reliable, because the allowable speed may have been determined to cover up manifold faults, or even carburetion faults. It can be stated, however, that for a manifold of a type giving good liquid distribution, which is properly heated, a gas speed not lower than 1000 ft p. m. should be allowed for the lowest speed at wide throttle.

With liquid distribution and exhaust heat, we will eliminate all our winter carburetion troubles, except the one of starting the cold engine. Hard starting however, is due more to the unadvertised weakness of the modern system of battery starting and ignition than to carburetion. We still have in the gasoline of to-day sufficient volatile components to permit the starting of the engine in very cold weather, provided the igniting spark is of sufficient intensity. The fact, however, is that in winter starting, when the hottest spark is needed, the ordinary ignition gives its weakest spark. The engine is hard to turn when very cold and the starting motor is called upon to deliver its full torque at a relatively low speed.

After the engine starts, we have means to "carburete" the much talked of heavy gasoline, but it must be done in the engine design as well as in the carbureter. The problem of cold weather starting is "up to" the manufacturer of starting, lighting and ignition apparatus to a very great extent, for if he cannot eliminate it, he can certainly alleviate the difficulty to such an extent that we will be able to start the engine as easily as years before.

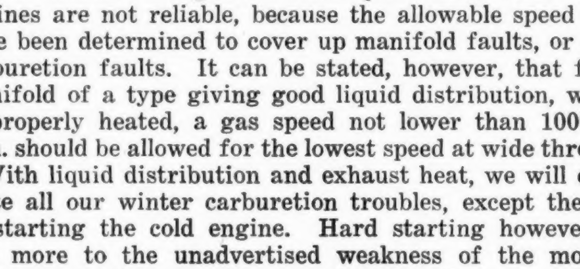


Fig. 12

Heat Treatment of the Alloys of Aluminum

This paper, translated from the French journal *L'Aeronautique*, shows that a striking similarity exists between the heat treatment of these alloys and the heat treatment of steels. Numerous tests have shown this, the results showing what may be accomplished in handling the metal.

CERTAIN alloys of aluminum possessing remarkable properties were developed considerably during the war. These alloys, generally known under the name of Duralumin, have a rather variable composition. One of these compositions is as follows:

Al, 93.9; Cu, 3.70; Mn, 0.61; Zn, 0.25;
Mg, 0.43; Si, 0.58; Fe, 0.53.

As used commercially this alloy tests as follows:
Tensile strength, 50,000—56,000 lb. p. sq. in.
Elastic limit, 18,000—24,000 lb. p. sq. in.
Elongation, 20—22 per cent.
Reduction in area, 35—38 per cent.
Resistance to shock, 4—5.5.
Brinell hardness (under 500 kg. with 10 mm. ball), 86—93.

These properties are due principally to the modifications in the metal by a special heat treatment. The metal, heated to 840 deg. F., is quenched in water and then left to stand a certain length of time. Quenching alone does not appreciably change the properties of the untreated metal.

Untreated metal:
Tensile strength, 38,000—42,000 lb. p. sq. in.
Elastic limit, 14,000—20,000 lb. p. sq. in.
Elongation, 20—22 per cent.
Reduction of area, 43—45 per cent.
Resistance to shock, 7—8.
Brinell hardness, 63—68.

Heat treated metal (immediately after quenching):
Tensile strength, 35,000—41,000 lb. p. sq. in.
Elastic limit, 14,000—17,000 lb. p. sq. in.
Elongation, 22—23 per cent.
Reduction of area, 41—45 per cent.
Resistance to shock, 8—9.
Reduction of area, 59—67 per cent.

But after a certain length of time there is a notable increase in tensile strength, elastic limit and hardness, with slight change in elongation and reduction in area as the characteristics show.

MM. Leon Guilbert, Jean Durand and Jean Galibourg have attempted to examine into this phenomenon by determining the existence of a point of transformation. They could not prove it by measuring the rates of cooling or expansion but they have determined the quenching temperature which must be reached in order to have a variation of the mechanical properties that take place after a certain period of aging.

Temperature at Quenching (Deg. F.)	Untreated Metal	Immediately After Quenching	Hardness After 1 Hour	After 24 Hours	After 48 Hours
570	65	61	61	61	61
750	65	61	61	61	61
840	65	61	61	65	93
930	65	61	61	86	93

Translated by Lieut. John Jay Ide, U.S.N.R.F.

The point of transformation occurs, therefore, between 750 and 840 deg. F. It will be noted, further, that within the range of temperatures given above the final hardness does not depend upon the temperature of quenching.

The theory which appeared the most probable is the following:

Quenching at temperatures above 750—840 deg. has the effect of maintaining the internal structure of the material obtained at high temperatures, but when quenching is done at ordinary temperatures a transformation takes place with a partial return toward the internal structure of the metal when cold.

Although it was not possible to put in evidence the difference in structure between the untreated and the tempered metal the following tests fully confirm the above theory. Now, since the hardening of the metal starts taking place as soon as an ordinary temperature is reached, the hardness of the metal would not change with time, if the metal was quenched and kept at a very low temperature after quenching. In fact, by quenching at 840 deg. in liquid air and keeping the metal in liquid air the following results have been obtained:

Un-treated Metal	Immediately After Quenching	Hardness After 24 Hr.	After 48 Hr.	After 72 Hr.	After 96 Hr.
65	61	61	61	61	61

It is to be noted that the metal reached its normal hardness of 93 at the end of 48 hours at 59 deg. after 96 hours of immersion in liquid air.

On the other hand, since the maximum hardness can be produced by a prolonged aging at ordinary temperatures, the same hardness should be produced more rapidly when the aging temperature is raised within certain limits.

The following figures clearly prove this hypothesis:

Aging Temperature After Quenching (Deg. F.)	Hardness after time of aging of: 0 Hr.	1 Hr.	6 Hr.	24 Hr.	48 Hr.
60	61	61	61	65	93
120	61	73	80	77	86
210	57	70	71	83	83
390	57	77	93	93	93
570	61	54	53	53	53

To sum up, maintaining the metal at a low temperature after quenching prevents it from hardening by aging; the maximum of hardness is attained more rapidly as the aging temperature approaches 390 deg.

We may thus conclude that (1) the heat treatment of alloys known as Duralumin causes the maintenance of the internal structure of the material obtained at high temperatures, and that (2) hardness is acquired by aging which can be carried on at ordinary temperatures, but the speed of which is increased by raising the temperature to not above 390 deg.

There thus exists a striking similarity between the heat treatment of this alloy and the heat treatment of steel.

The Design of Cooling Surfaces for Air-Cooled Engines

The factors affecting the efficiency of fins for air cooled engines are developed in this article, with many references to other authorities. The equations and tables given here should be very useful to engineers.

By W. Byron Brown*

IN the case of air-cooled engines, most of the cooling is accomplished by means of metal fins, usually circumferential, but sometimes, as in the case of the Franklin automobile engine, longitudinal. In designing fins for the cooling of such a cylinder, three main questions arise:

1. How much cooling surface is required?
2. What is the best shape of fin to use?
3. What spacing between fins should be adopted?

This paper gives some new formulæ and tables which partially answer these questions.

The answer to the question of the necessary cooling surface requires a knowledge of the amount of heat to be dissipated, the air speed available for the cooling fins, and the allowable cylinder temperature.

Measurements made in England on the heat dissipated from the cooling surfaces of air-cooled engines show that this is about 6 per cent greater than the brake horsepower.†

The rate of heat dissipation from the surface of an air-cooled cylinder, i.e., the heat dissipated from unit surface per unit time per unit temperature difference between the air and the surface has been measured in a few cases. It has been found to be roughly proportional to the air density, temperature difference between cylinder wall and air, and the air speed. A closer relation is,

$$q \sim (\rho V)^{0.8}$$

between 15 and 120 miles per hour,

where $q =$ rate of heat dissipation
$\rho =$ air density
$V =$ air speed in miles per hour.

The same experiments showed that q in c.g.s. units is about 0.005 at 50 miles per hour.

Let $\theta =$ average temperature difference between cylinder wall and air in deg. Fahr.
$Q =$ heat dissipated in horsepower.
$S =$ equivalent surface in square feet.

By equivalent surface is meant surface that dissipates heat at the same rate as cylinder wall surface. To express this in an equation a new factor is needed. This will be called the effectiveness of the cooling surface and is the ratio of the heat dissipated by the surface to that dissipated by the *same* area of adjacent cylinder wall surface. That is, if 1 sq. ft. of fin surface dissipates 1 hp. and 1 sq. ft. of cylinder wall surface adjacent (so as to have the same temperature as the fin base) dissipates 2 hp., then the effectiveness is

1 hp. divided by 2 hp., or 50 per cent.

If now $C =$ cylinder wall surface in square feet,
$F =$ fin surface in square feet,
and $f =$ effectiveness of fin surface,

then $S = C + fF$ ....(1)

It is shown in Appendix 2 that the effectiveness of correctly proportioned fins is 63 per cent, so that (1) may be written for this case

$$S = C + 0.63\,F \quad \text{......................(2)}$$

If $q = 0.005$ is expressed in the English units used above, that is, as $\dfrac{HP.}{ft.^2\,deg.\,F.}$, the result is $\dfrac{1}{70}$.

Therefore

$$q = \frac{1}{70} \text{ at 50 miles per hour.}$$

$$q = \frac{V}{3500}\,\frac{HP.}{ft.^2\,deg.\,F.}$$

$$\text{and } Q = \frac{V S \theta}{3500} HP.$$

Solving this for S gives

$$S = \frac{3500\,Q}{V\theta}\,ft.^2 \quad \text{...............(3)}$$

The best working temperatures for air-cooled cylinders have been measured by A. H. Gibson,‡ who finds that the maximum temperature of the head should not exceed 270 deg. C. (about 520 deg. Fahr.), and that between this temperature and 200 deg. C. (about 400 deg. Fahr.) results are as good as with water-cooled engines of similar design and size. If the maximum temperature above the cooling air is 430 deg. Fahr., the mean cylinder wall temperature is only about 270 deg. Fahr.

Assuming, then, a mean temperature difference of 250 deg. Fahr., equation (3) gives the following table:

TABLE I

Air Speed, mi. per hr.	Equivalent Cooling Surface sq. ft. per B. H. P.
30	0.47
40	0.35
50	0.28
60	0.23
70	0.20

*Of the Bureau of Standards.

†*Experiments at the Bureau of Standards indicate a rather smaller rate of heat dissipation than this.*

‡*Theories and Practices in the Air Cooling of Engines,* AUTOMOTIVE INDUSTRIES, *May* 13, 1920, *p.* 1109.

A fin with slightly concave surfaces and having a sharp tip is given by Gibson as the best shape, but a plain wedge-shaped one of the shape shown in Fig. 1 is very nearly as good.

It is shown in Appendix 2, that

if w = fin length,

t = average fin thickness,

k = metal conductivity,

q = rate of heat dissipation, all in consistent units, the proportions that give minimum fin weight for a

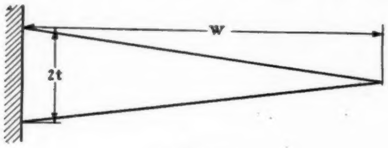


Fig. 1

given heat dissipation are given by equation (4).

$$\frac{w^2}{t} = \frac{k}{q} \quad \text{(4)}$$

Table II for the case where $V = 40$ miles per hour has been published by Gibson. The metals used had conductivities of 0.38, 0.10 and 0.90 c.g.s. units for aluminum alloy, steel and copper, respectively.

TABLE II.

Bottom Breadth, cm.	0.025	0.05	0.1	0.2	0.3	0.4	0.5
Length, cm.							
Aluminum	....	....	2.0	2.9	3.5	4.1	4.5
Steel	....	....	1.1	1.5	1.8	2.1	2.3
Copper	1.6	2.3	3.3	4.8	...	...	...

Table III shows the values given by equation (4) for the same face. The average fin thickness is here half the bottom breadth, of course.

TABLE III

Bottom Breadth, cm.	0.025	0.05	0.1	0.2	0.3	0.4	0.5
Avg. Thickness	0.0125	0.025	0.5	.10	.15	.20	.25
Length, cm.							
Aluminum	1.1	1.5	2.1	3.0	3.6	4.2	4.7
Steel	0.54	0.76	1.1	1.5	1.8	2.1	2.3
Copper	1.6	2.3	3.3	4.6	5.6	...	...

The agreement is seen to be very good.

The cases where the fins are not wedge-shaped but have blunt edges can be handled easily by a slight modification of equation (4). Suppose the fin has a shape like Fig. 2.

It can be shown that if b is the fin thickness at the exposed edge and $w' = w + b/2$, and if t is the average fin thickness, equation (4) becomes

$$\frac{w'^2}{t} = \frac{k}{q} \quad \text{(5)}$$

TABLE IV.
Fin Dimensions for Aluminum, Steel and Copper at Different Air Speeds.

Aluminum Alloy, k = 0.38

Length, in.	0.4	0.6	0.8	1.0	1.25	1.5
Thickness, in.						
30 m.p.h. (q = 0.003)	0.0032	0.0072	0.013	0.020	0.031	0.045
50 m.p.h. (q = 0.005)	0.0053	0.012	0.021	0.033	0.052	0.075
70 m.p.h. (q = 0.007)	0.0075	0.017	0.030	0.047	0.073	0.105

Steel, k = 0.010

Length, in.	0.4	0.6	0.8	1.0	1.25	1.5
Thickness, in.						
30 m.p.h. (q = 0.003)	0.012	0.027	0.049	0.076	0.12	0.17
50 m.p.h. (q = 0.005)	0.020	0.046	0.081	0.127	0.20	0.29
70 m.p.h. (q = 0.007)	0.028	0.064	0.11	0.18	0.28	0.40

Copper, k = 0.90

Length, in.	0.6	0.8	1.0	1.25	1.50	1.75
Thickness, in.						
30 m.p.h. (q = 0.003)	0.0030	0.0054	0.0085	0.013	0.019	0.026
50 m.p.h. (q = 0.005)	0.0051	0.0090	0.014	0.022	0.032	0.043
70 m.p.h. (q = 0.007)	0.0071	0.013	0.020	0.031	0.044	0.060

The difference between the efficiency of the two extreme cases, i.e., rectangular fins where $b = t$ and wedge-shaped where $b = 0$, is about 8 per cent in the case considered, where proportions are given by equation (5), so that equation (5) holds within these limits for any such shape.

If w' and t are expressed in inches and k and q in c.g.s. units, (5) becomes

$$\frac{w'^2}{t} = 0.394 \frac{k}{q} \quad \text{(6)}$$

and tables are easily prepared which give the correct thickness (i.e., ½ the base thickness) for fins of several lengths when made of different metals and used at different air speeds.

Fin Spacing

From considerations of weight only, it is easily seen that the smaller spacing between fins is the better. For instance, let the spacing be halved, then the same cooling surface may be obtained by halving the fin length (approx.). But since the fin thickness by equation (5) varies as the *square* of the fin length, one-quarter of the fin thickness will suffice. The fin weight is proportional to the fin volume, which is the product of the surface and the thickness. Since the surface is unchanged and the thickness is one-quarter, the weight will be one-quarter also.

This reasoning is applicable for wide spacings where a reduction in spacing does not seriously impede the air flow. But decreasing the spacing below certain limits does not pay because the air flow over the inner fin surfaces is reduced too much for good cooling.

In the case of radiators, it is known* that decreasing the air spaces below ¼ in. generally fails to increase the cooling power because the air flow through the radiator is reduced faster than the cooling surface is increased. The same thing is true of air-colled cylinders, and the experiments of Gibson indicate that this limit is about ¼ in. for machined steel cylinders and 5/16 in. for cast-iron and aluminum alloy cylinders.

Fig. 2

The principal conclusions may be summarized as follows:

1. To secure the best cooling for minimum weight, the width of air space between fins should be about ¼ or 5/16 in.
2. The best fin shape is approximately wedge-shaped, the proportions being given by Table IV, and by equation (6) used in preparing it.
3. The equivalent cooling surface is given by Table I, and by equation (3) used in computing it, and the fin surface required by equation (2).

APPENDIX I.

Effectiveness of a Fin

This will be given here for a simple case. The more general case requires considerable mathematics which confirms this result.

*"*Aeronautical Radiator," Aerial Age Weekly, March 3, 1919, p. 1261.*

Fin effectiveness is defined as the following ratio:

$$\frac{\text{Heat lost by the fin per unit time.}}{\text{Heat lost by the same area of wall surface per unit time.}}$$

Consider a section of a long rectangular fin with constant base temperature so that heat flows straight out from the walls. The temperature across the thickness of the fin is assumed constant, from the fact that the conductivity is from 20 to 180 times the rate of transfer from the surface.

It is assumed that the rate of heat dissipation is uniform from the tip to the root of the fin. This is not quite true, for a British experimenter found a variation in air flow of something over 15 per cent, but by using an average value, an idea of the form of function may be gained.

Let Fig. 3 be a section of the fin.

This sort of fin will lose some heat from the exposed edge. It will simplify the mathematics considerably to imagine the fin extended by a distance equal to half the fin thickness and losing no heat from the exposed end. Analysis shows that the effectiveness computed in this way differs from the true effectiveness by less than 0.05 per cent.

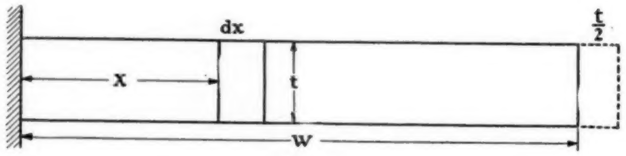


Fig. 3

Let q = coefficient of heat transfer from the fin surface,
w = true length of fin,
w' = corrected length of fin = $w + t/2$,
t = fin thickness,
x = distance from the cylinder wall,
f = fin efficiency,
k = fin conductivity,
θ = temperature of fin above the air,
θ_o = temperature of cylinder wall above the air,
H = heat dissipated by the fin per unit time.

Unit depth perpendicular to the plane of the figure is assumed throughout.

The cooling surface of the fin is $2w + t$, therefore the heat dissipated by cylinder wall adjacent of same area is

$$2q\,\theta_o\,(w + t/2) = 2q\,\theta_o\,w'$$

$$\text{Hence } f = \frac{H}{2q\,\theta_o\,w'} \qquad \text{.............. (1)}$$

$$\text{and } H = 2\int_o^{w'} q.\,dx.\,\theta \qquad \text{........... (2)}$$

$$f = \frac{1}{w'\,\theta_o}\int_o^{w'} \theta.\,dx \qquad \text{.............. (3)}$$

The heat conducted into an element dx is $kt\dfrac{d\,\theta}{dx}$

while that conducted out is $kt\left[\dfrac{d\,\theta}{dx} + \dfrac{d}{dx}\left(\dfrac{d\,\theta}{dx}\right)dx\right]$

The difference between these two is $kt\;\dfrac{d^2\,\theta}{dx^2}\,dx$, and is, of course, the heat lost from the two surfaces, which is $q.2\,dx\,\theta$.

$$\therefore\; kt\,\frac{d^2\,\theta}{dx^2}dx = 2q\,dx\,\theta \qquad \text{.............. (4)}$$

or

$$\frac{d^2\,\theta}{dx^2} = \frac{2q}{kt}\,\theta \qquad \text{................. (5)}$$

Let $a^2 = \dfrac{2q}{kt}$ Then a solution of (5) is

$$\theta = A\cosh\,(ax - B) \qquad \text{............. (6)}$$

where $\theta = \theta_o$ when $x = O$

and $\dfrac{d\theta}{dx} = O$ when $x = w'$ .... (7)

Therefore θ_o = A cosh B......................... (8)

$$O = Aa\sinh\,(aw' - B) \qquad \text{................. (9)}$$

(9) can only be satisfied by $B = aw'$

$$\therefore\; A = \frac{\theta_o}{\cosh aw'} \qquad \text{............... (10)}$$

$$\text{and } \theta = \frac{\theta_o}{\cosh a\,w'}\cosh\,(a\,x - a\,w') \qquad \text{............ (11)}$$

Putting this value of θ into (3) gives

$$f = \frac{1}{w'\cosh aw'}\int_o^{w'}\cosh a\,(x - w')\,dx \qquad \text{....... (12)}$$

$$f = \frac{\sinh aw'}{aw'\cosh aw'} \qquad \text{.......................... (13)}$$

$$= \frac{\tanh aw'}{aw'} \qquad \text{........................... (14)}$$

It is possible by the use of Fourier Series to prove that (14) is true for any temperature distribution whatever along the fin base. It is rather involved to give in full here.

The case of a circumferential fin gives f in terms of Bessel functions of an imaginary quantity. Owing to the spreading along a radius, f is about 5 per cent less than given by (14) for cylinder diameters of 3 to 6-in. and proportions of equation (6).

An analysis of the case of tapering fins gives f in terms of Bessel functions also. In this case f is about 6 per cent greater than (14) gives. If then, the fins are both sloping and circumferential, (14) gives a result not far from the truth.

APPENDIX II.

Condition for Least Weight

This requires the adjustment of w' and t so that the equivalent surface $S = f\,F$ shall be a maximum for a given fin weight.

Let F = fin surface
M = weight of one fin
s = fin density

By (14)

$$S = f\,F = \frac{2\,w'\tanh aw'}{a\,w'} \qquad \text{........... (15)}$$

and since $a^2 = \dfrac{2q}{kt}$

$$S = \frac{2}{\sqrt{\dfrac{2q'}{kt}}}\tanh w'\sqrt{\frac{2q}{kt}} \qquad \text{.......... (16)}$$

The fin weight

$$M = s\,wt$$

For a first approximation let w' be substituted for w, since they are nearly equal.

Then

$$S = \frac{2}{\sqrt{\frac{2q}{kt}}}\tanh\frac{M}{st}\sqrt{\frac{2q}{kt}} \quad \ldots\ldots\ldots (18)$$

The only variables are S and t. The problem is to find the value of t that will render S a maximum. This is done in the usual way by differentiating (18) with respect to t and equating dS/dt to zero. The result is the equation below when for M has been substituted its value swt.

$$3\,w'\sqrt{\frac{2q}{kt}}\ \mathrm{sech}^2\, w'\sqrt{\frac{2q}{kt}} = \tanh w'\sqrt{\frac{2q}{kt}} \quad \ldots\ldots (19)$$

This equation is of the form

$$3\,y\ \mathrm{sech}^2\, y = \tanh\, y \quad \ldots\ldots\ldots\ldots\ldots\ldots (20)$$

and may be solved graphically or by trial.

There are two solutions

$$\left.\begin{aligned} y&=0 \\ \text{or } y&=1.42. \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots (21)$$

that is

$$w'\sqrt{\frac{2q}{kt}} = 1.42 \quad \ldots\ldots\ldots\ldots\ldots\ldots (22)$$

The first is of no use in this problem. The second is, for it gives the relation between w' and t that will make S a maximum. Since 1.42 is nearly equal to $\sqrt{2}$, (22) may be written

$$\frac{w'^2}{t} = \frac{k}{q} \quad \ldots\ldots\ldots\ldots\ldots\ldots (23)$$

If the approximation $w'=w$ is not made, (20) becomes

$$\tanh\, y = 3y\ \mathrm{sech}^2 y\left(1 - \frac{5t}{6w'}\right) \quad \ldots\ldots\ldots (24)$$

If t is as much as 10 per cent of w', the result in (22) is 1.38, or 3 per cent less than before. The error is generally much less than this.

In Appendix (I) it was shown that the fin efficiency

$$f = \frac{\tanh aw'}{aw'} = \frac{\tanh w'\sqrt{\frac{2q}{kt}}}{w'\sqrt{\frac{2q}{kt}}}$$

For the case where S is a maximum, (22) shows that

$$f = \frac{\tanh 1.42}{1.42} = 63 \text{ per cent.}$$

Thus the efficiency of fins that are proportioned to have maximum equivalent cooling surface for a given weight is always 63 per cent for all materials and air speeds.

Heat Treatment of High Chromium Steels

An investigation of the effect of heat treatment on the tensile properties, hardness and micro structure of high chromium steel (stainless steel) has been made by the Bureau of Standards. Following is the chemical analysis of the samples on which the tests were made:

Carbon, 0.29 per cent.
Manganese, 0.38 per cent.
Silicon, 0.70 per cent.
Chromium, 13.2 per cent.

When samples of this steel are quenched in oil from various temperatures, it is found—

(a) That hardness, as measured by Brinell and Shore instruments, increases with increasing quenching temperature until a temperature of about 1950 deg. Fahr. is reached. Maximum range of hardness is generally obtained by quenching from this temperature, up to the highest heat used, but in some cases this hardness actually decreases, due to retention of the solid solution.

(b) That quenching from about 1750 deg. Fahr. develops the best combination of strength and ductility which is not coincident with range of maximum hardness. Quenching from this or lower temperature does not retain all the carbide in solution, as is the case in samples quenched from considerably higher temperatures, notably 2100 and 2250 deg. Fahr.

(c) That ductility as measured by elongation and reduction is very low in those samples quenched from 1850 deg. Fahr. or above.

Short-time tempering at temperatures up to about 800 deg. Fahr. of samples previously quenched from both 1750 and 2100 deg. Fahr. decreases brittleness. However, ductility is increased to a greater extent in those samples quenched from 1750 deg. Fahr. than those quenched from the higher temperatures. Tempering above about 800 deg. Fahr. markedly decreases strength values and hardness, which is, of course, accompanied by greatly increased ductility. In general, the structure of the hardened steel tends to persist even when tempered for a short period of time at temperatures comparatively close to the lower critical range, the characteristics depending upon the quenching temperature used. The most rapid change in tensile properties and hardness occurs in tempering between about 800 and 1000 deg. Fahr.

KMCO Bearing Metal

It has been known for a long time that lead possessed very good self-lubricating qualities, but its other properties have prevented its use as a bearing metal. A new bearing metal with lead as a base has been put on the market by H. Kramer & Co. and is known as "KMCO." It is an alloy of copper, tin and antimony with lead. The metal has a definite melting point of 680 deg. and a Brinell hardness figure of 28.4. It is claimed that this metal will melt as an alloy and not as a physical mixture of metals and, furthermore, will not crystallize. The hardness of this metal does not fall off as rapidly as that of babbitt, the Brinell hardness figure being 21.1 at 212 deg. A test of this metal was made without lubricant at a pressure of 100 lb. per sq. in. and 650 r.p.m. The metal did not flow for thirty minutes and when it did it adhered to the shaft and was afterward easily removed. The price is considerably less than the usual bearing metal price. Several large companies have already tested this metal and have adopted it for use in quite large quantities, we are informed. Tests with bearing pressures up to 800 lb. per sq. in. and 600 r.p.m. are said to have failed to develop excessive bearing temperatures.

The Theory and Practice of Regulating the Generator Output

The different car speeds would cause the generator to deliver either too little or too much current. This article gives some of the advantages of one type of regulation commonly employed to-day. A very useful résumé.

By J. T. Fitzsimmons*

THE success of the equipment on a motor car depends to a great extent upon its ability to give unfailing performance with minimum attention. Any device which is rugged and devoid of parts requiring delicate adjustment has much in its favor. This is especially true of electrical equipment on the more popular priced automobiles, where frequent inspections by the authorized representatives of the manufacturer are not feasible.

The electrical generator having the third brush method of regulation is a development along these lines, and the fact that it is used on at least 75 per cent of the motor cars built to-day is ample proof that it has given most satisfactory service.

For the sake of simplicity, modern design demands that the generator be driven at a certain fixed gear ratio to the motor. This gear ratio varies from 1:1 on generators designed for low speeds, to 3:1 on those which are intended to give satisfactory performance at higher speeds. On a generator having a gear ratio of 1½:1, the speed may rise to 4500 r.p.m. at 60 m.p.h. car speed, while at speeds corresponding to 10 m.p.h., the r.p.m. will be only 750. In order to have a really good design, such a generator should begin to charge the battery at not less than 650 r.p.m., and at about 1000 r.p.m., corresponding to 13 m.p.h. car speed, should have sufficient output to take care of the lighting and ignition load, which will be in the neighborhood of 8 amperes. Such a generator, on the average car, should at no time have a maximum output of more than 20 amperes; otherwise injurious heating of this unit or the battery will result. These high charging rates, if continued any length of time, are especially injurious to the storage battery, since the present tendency is to keep the battery small and light, so that it has little excess capacity for taking care of emergency conditions. It is therefore desirable to have a generator unit which has a higher charge rate at low speeds than at high, since short runs giving a high charge rate—as would be the case in city driving—would not injure the battery, but at the same time would give ample input to take care of frequent cranking. In touring, where relatively high speeds may be maintained for a considerable time, the charging rate will be low enough to protect the battery. It is also very probable that fewer starts will be necessary in touring; therefore, less input to the battery will be necessary, for, after all, electrical cranking is the big battery load. The generator with the third-brush method of regulation practically meets these conditions, and, although it is possible to build regulating devices more theoretically correct, as yet there has been nothing produced commercially which is nearly so rugged and which requires so little care.

Engineer, Research Department, The Dayton Engineering Laboratories Co.

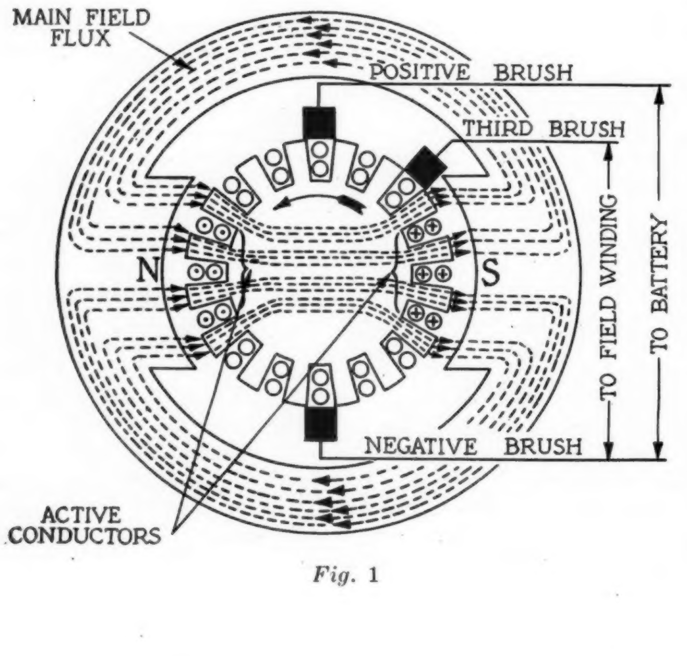


Fig. 1

Regulation Due to Armature Reaction

The regulation of the output of a variable-speed generator, by means of a third brush, to which is connected one terminal of the field winding, is due to armature reaction. This is the effect of the magnetic field set up by the flow of current in the armature, upon the magnetic field set up through the poles of the machine, by the flow of current through the field winding. This effect depends upon the armature current, and the third brush system is consequently a current-regulating system.

The voltage generated in the armature of any generator depends upon the number of conductors on the armature and the strength of the magnetism or magnetic flux set up by the field. Numerically. it is equal to the product of the magnetic flux entering each positive pole, the number of poles, the total conductors on the armature and the number of revolutions per second, all divided by 100,000,000 times the number of paths in parallel between brushes of the armature.

In a variable-speed automobile generator, it is essential that the generator voltage does not vary more than 25 or 30 per cent from the open-circuit voltage of the battery, regardless of speed; otherwise the charge rate on the battery becomes too high and trouble is experienced with the excessive voltage on the lamps. From the statement just made, it will be seen that the number of poles, the number of paths in parallel in the armature and the number of conductors in the armature are fixed quantities

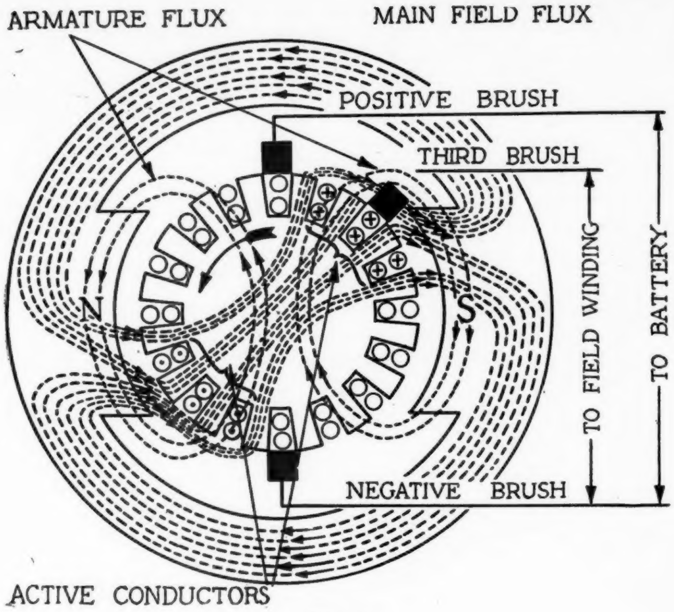


Fig. 2

worked out in the design of the machine, and cannot well be changed for regulating purposes. The speed varies from approximately 11 r.p.s. to 75 r.p.s., or increases almost 700 per cent with but 30 per cent allowable increase in voltage. All quantities except the field flux and the speed being fixed, to keep the voltage nearly constant, it is necessary to decrease the field flux as the speed increases. This means a reduction of field current, or a reduction of voltage across the field winding, which is accomplished by the third-brush method of connection.

In Fig. 1 is shown a diagram of a two-pole generator with conductors shown in slots in the armature, the brushes as indicated and the positive and negative field poles represented as *N* and *S* respectively. When no current flows in the armature, the field flux takes the path indicated by the arrow heads. In the conductors there is generated a voltage, the direction of which is indicated by the crosses and dots, the former meaning from front to back, or from the observer, the latter toward the observer. Each conductor may be considered as linked up in parallel with one on the armature directly opposite it, and in series with the one in the same slot. Since the voltage in the armature depends upon the speed with which the lines of magnetic flux are cut, at the particular armature position shown in Fig. 1 the conductors midway between the positive and negative brushes have a maximum voltage generated in them, due to their motion at right angles to the field flux, while the conductors under and near the positive and negative brushes have little or no voltage generated in them, since they are traveling parallel to the field flux. The field winding is connected between the third or regulating brush and the negative brush, and under the conditions illustrated in Fig. 1, the voltage across these brushes is practically the same as that across the positive and negative brushes. This is due to the fact that the conductors between the third and the negative brush are the same as between the main brushes, with the exception of those in the zone of the armature from the third to the positive brushes. These at this time are inactive, however, so no loss in this type connection results, and the field strength, with but little current flowing in the armature, is practically the same as if the field winding were connected across the two main brushes.

When a current flows in the armature, different conditions result. These are shown in Fig. 2, and are due to the magnetic field set up by the flow of current in the armature, in a direction so as to cross the main field flux as shown. The net result of this is to distort the main flux as indicated. The active conductors have now shifted in the direction of armature rotation. The voltage across the regulating and the negative brushes is no longer the same as across the main brushes, since the number of active conductors cutting the main flux is now the same as in Fig. 1, notwithstanding the fact that they are now different conductors. The active conductors between the regulator and the negative brush have been reduced, however, by this shift, and the voltage becomes proportionally less. This drop in voltage across the field windings means diminution of the main field strength and a reduction of charge rate. Any reduction in main field flux necessarily means that the effect on it, resulting from the flux set up by the armature, becomes proportionally larger, and a condition finally arrives where a balance is established. In both Figs. 3 and 4 it must be understood that the generators shown are diagrams only used to illustrate the conditions just indicated. In the actual machine, the inactive conductors on the armature are relatively few, since the polar span covers a large proportion of the armature.

From the preceding, the effect of the position of the regulating brush with respect to the main brushes is readily seen. When the third brush is located near the positive brush, there is very little regulating effect, since the flux between the third and negative brush is practically the same as between the main brushes. Under these circumstances the charge curve plotted to speed will tend to flatten out somewhat with output, as shown in Fig. 3, due to brush loss, resistance loss in the armature, iron losses and the effect of the field distortion upon the armature conductors. The last becomes so great at times as to cause a reversal of voltage in some of the conductors between brushes. Under these circumstances commutation on the main brushes may become rather bad, due to the voltage generated in the armature coils being short-circuited by the brushes. At such a time, if the voltage were measured between opposite commutator bars on a two-pole machine, it would be found that a certain point could be found where the voltage was considerably higher than across the main brushes.

With the third brush set some distance away from the

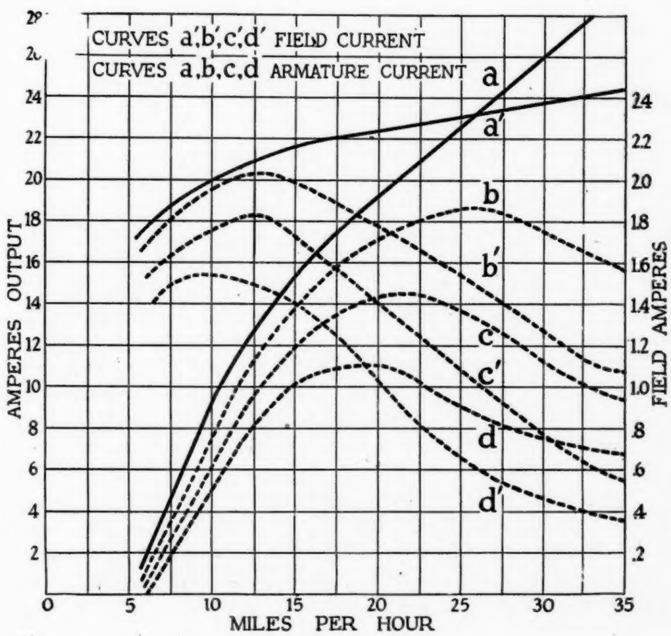


Fig. 3

positive brush, a very marked regulating effect will be noted, since a very slight armature current will produce enough shift of field flux to cause the voltage to drop across the third and the negative brush. If the third brush is set too far away from the positive brush, the voltage across the third and the negative brushes will always be less than across the main brushes, resulting in the generator having a higher speed when it begins to charge the battery than would normally be the case. This is sacrificing a condition for which the engineers who design these machines most strive.

Output-Speed Curves

In Fig. 3 is shown output in amperes plotted against speed, with three different third-brush settings, all of which were within the range of adjustment provided by the manufacturer. There is also shown the output with field winding connected across the main brushes and the amperes flowing in the field winding in all these cases. Curves *a* and *a'* show conditions without regulation, *bb'*, *cc'* and *dd'* with third-brush settings to give high, average and low charge rates respectively. All of these curves are characteristic of a two-pole machine, thousands of which have given satisfactory service.

Fig. 4 shows the variation of voltage with output, across the main brushes as well as across the third and positive brush, the generator having a constant excitation from an outside source. All curves give results with the generator connected to a three-cell battery. The effect of field distortion upon the voltage across the shunt field, when connected to a third brush and to a main brush of opposite polarity, is clearly shown. The voltage at this point gradually decreases with the generator output, until it falls to less than one-half the voltage across the main brushes. From Fig. 2 it can be seen that under these conditions over one-half the flux is passing through the armature between the positions indicated by the positive and the regulating brush, due to the crowding of flux to the pole tips.

As has already been set forth, therefore, the regulation of this type of machine is inherent and not subject to easy disarrangement. Adjustment of charge rates to take care of different driving conditions are easily and surely made without disturbing the operation of the unit. The satisfactory performance of the machine is fundamentally a question of proper design, and a few facts relating to this are worth while.

Proper Design of Third Brush Generators

It so happens that the armature conductors responsible for field distortion are those opposite the pole faces. The reluctance of the magnetic circuit in which is established this magnetic field set up by the armature, depends upon the air gap between the armature and the fields. In constant-speed generators, where close regulation is desirable, it is customary to keep the polar embrace rather low and the air gap wide, to reduce armature effects. Where large armature reactions are desirable, as in variable-speed generators of the third-brush type, it is an advantage to make the air gap small and the polar embrace high. These conditions also permit a reasonable output at very low speeds, since the full strength of the field and practically all the conductors of the armature may be utilized.

Commutation is a very important item on all direct-current machines. To secure ideal results it is necessary to have the armature coils, at the time they are undergoing commutation, in a very weak field, since at this time they are short-circuited by the brushes, and any voltage generated in them may cause serious arcing. It is impossible to do this on machines in which the field is constantly shifting, due to armature reaction. The only way of meeting this condition, therefore, is to keep the voltage generated in each coil low by reducing the number of turns as much as possible, resulting in commutators with a very low voltage drop per bar. Likewise the brushes should be as narrow as possible compatible with sufficient area of contact, since too high contact resistance, although assisting commutation, may cut down the output of the machine too much below normal. The regulator brush should also be as narrow as possible, in order to cut down the short-circuit current in the coils shorted by it, since these coils are nearly always cutting a relatively strong magnetic field. For these reasons, brush material must be carefully selected and proven by extensive tests, and great care should be taken, in case brush replacements are necessary, to use only those grades found satisfactory in the machine to which they are applied.

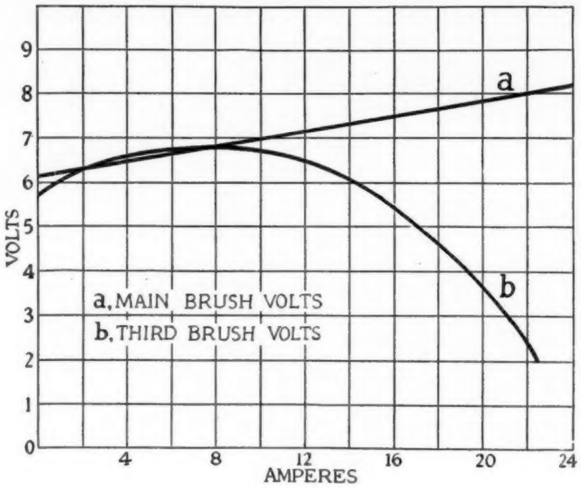


Fig. 4

Two-Pole Generator for Medium-Priced Cars

For installations in medium-priced cars, the two-pole machine will probably predominate, since the greater distance between pole centers, as well as between main brushes, allows greater range of regulator brush adjustment and consequently greater ease in manufacture. For high-class cars, however, where price is not of major importance and close manufacturing limits may be held, some of the present four-pole machines show marked advantage in size, performance and efficiency.

To repeat previous statements, with the design properly worked out, the generator with third-brush regulation is practically trouble-proof, as far as that feature of the machine is concerned, and has certainly added much to the reliability of the electrical equipment of the modern motor car.

Quantity Motorcycle Production

THE Pullin motorcycle recently brought out in England has many features facilitating quantity production. The frame is of pressed steel and contains all the parts. It is supported on the wheels by telescoping tubes. The engine is a horizontal, two-stroke, air-cooled, single-cylinder. Control is had by varying the amount of mixture returned to the crankcase on each stroke. There are two speeds given by an epicyclic gearset controlled from the handle bar grip. Chain drive is employed and two automobile type brakes. Lighting and ignition is taken from a flywheel magneto. The price is about $250, normal exchange.

English Comment on the American Made Automobile

This article is a review of a series of criticisms published by the British writer upon the design and performance of cars manufactured in this country. American engineers and designers should be interested in his conclusions as they reflect European ideas upon such points.

A SERIES of criticisms of American made motor cars is being published in the London *Times*, one of the most influential of the British dailies. This paper takes rather exceptional interest in the automobile, and it is now particularly concerned with the "invasion" of the English markets by automotive products made in this country. The criticisms are being written by "Our Motoring Correspondent," who states that he has driven American cars both in the United States and in Europe. His conclusions appear typically British, and their trend may be more or less anticipated by anyone at all familiar with European cars and their construction. Nevertheless, the articles reveal something of the British attitude toward the American product.

Great numbers of automobiles made in this country are being shipped to England and to the English colonies; in fact, the English-speaking territories of the globe take first rank as the foreign markets of our automotive products. Exports of cars to these countries have been mounting month by month, despite the possibilities of increased British production. Prospects of increased British production are being made the foundation of appeals to colonial buyers to hold off from purchasing American cars, and, therefore, some attention should be given to English criticism of American design, not only by sales organizations but by engineers and designers as well.

The *Times* critic finds, first of all, that the American made car is a "family machine," few of them being "really fast." Most American cars are uncomfortable at speeds higher than 40 to 50 m.p.h., the figure varying with the car.

"American cars—even the very best and most expensive—are not always noted for very good engine balance at high speeds," the critic says. "Few that have come under the notice of the writer are capable of being driven all out without a certain amount of vibration and thrash at anything over 50 m.p.h. The degree varies in different engines and in different types of engine, but to a certain extent it is to be found in most of them."

In another of the articles, the writer says:

"Very few American cars are really fast. Some of them are capable of high bursts of speed on occasion, but it is noticeable, as a rule, that the smooth running of the engine falls off rapidly after a speed of 40 m.p.h. has been reached. British and Continental cars of the best kind are nearly all designed to maintain the same level of sweet running at any and all speeds. Motorists demand a good deal of speed nowadays, whether they use it often or not, and a car which, admirable in other ways, cannot bring its speed indicator to the 50 mark without harshness is not likely to be a commercial success."

Despite these features, the writer has found much that is praiseworthy in American automobiles. He finds them to be machines that will keep up their average performance day after day, and their average performance is rated as high. Contrasting the American idea of touring with that prevailing in much of Europe—the car being largely for sporting uses and driven frequently at speeds that would be universally condemned in the United States—the *Times* seems to have made a good case for the American product as a safe, reliable and economical machine.

"The good American car scores on the question of averages," the writer states. "The engines would begin to show symptoms of thrash, things generally would begin to vibrate, and the comfort of passengers and driver alike diminished in direct ratio to the increase of speed. On the other hand, these cars have usually shone at a keeping to a high average, such as 30 m.p.h., without unnecsary noise and with great ease and comfort of driving. Their comparatively big engines and low compression and revolution rates have combined to produce a high level of flexibility.

"That is what is meant by the description family car. A good American machine is one which seems pre-eminently sober, not to say sedate. It has few spectacular features and will, in all probability, be handsomely beaten, engine for engine, by any equally good European car, both in hill-climbing and on the level. Yet, at the end of a long day's run over give-and-take roads, you will probably see the American car close behind its more sporting competitor."

One car is criticized because the engine is not "truly accessible," another because of the black cast cylinders, and difficulty is found in one model with the steering apparatus, the latter difficulty, however, being said to be rare.

These remarks of the *Times* writer are of value to the American engineer and designer only because they single out the points that our European competitors are endeavoring to make. Combating them is, perhaps, primarily the job of the sales and advertising departments, but they should be considered, nevertheless, by the engineer who is responsible for the car. This is particularly true in regard to automobiles designed for export or for use in competition with the products of Europe. Most likely the *Times*' critic is voicing not only his own "pet" complaints but those of British engineers and car builders, and that is the main reason for reproducing them here.

One other point criticized in the series is in connection with body and coach work. European makers long have prided themselves upon this part of their work and complaints concerning the American bodies are not surprising. But it is of value to know just what is being said of this feature.

"The body is roomy and comfortable enough behind. As high-class American bodies go, the ——— five-seater is good, except in so far as the driver is concerned. His seat is really unpardonably cramped. It is not a question of elbow-room. In this case, there is ample width in the front seat."

Design and Structure of the German Metal Airplane

The Junker machine, as this military production was known, has been a source of interest and curiosity to engineers and designers who recognize, of course, the great production value of such construction. This description of the German plane is based upon studies by the British air service.

DURING the latter part of the war much effort was being devoted in all the chief belligerent countries to the problem of all-metal airplane construction. There were several reasons which made the solution of this problem seem very important. In the first place, the woods specially suited for aircraft building, such as spruce, were getting to be very scarce and, in any case, were difficult to deliver at the aircraft factories because they grow chiefly in dense forests in remote districts. Secondly, wood, wire and canvas construction does not lend itself to the same or similar processes of mass production as metal construction. Thirdly, an all-metal airplane is a fireproof machine and, therefore, much safer in war than one of the conventional type.

The Germans, toward the end of the war, used an armored machine known as the Junker, in which the wings, body, etc., were entirely of metal. One of these planes was captured by the British, and the British Air Service some time ago issued a rather lengthy report with numerous illustrations detailing the construction of this new design.

The Junker J-1 is radically different from the usual type of airplane, whether considered from the point of view of design or of actual construction. It is evidently a serious attempt to reduce to a minimum the dangers due to enemy action and to lengthen the life of the machine under exposure to rough weather and bad handling. To this end the machine is armored and all vulnerable parts,

Fig. 1

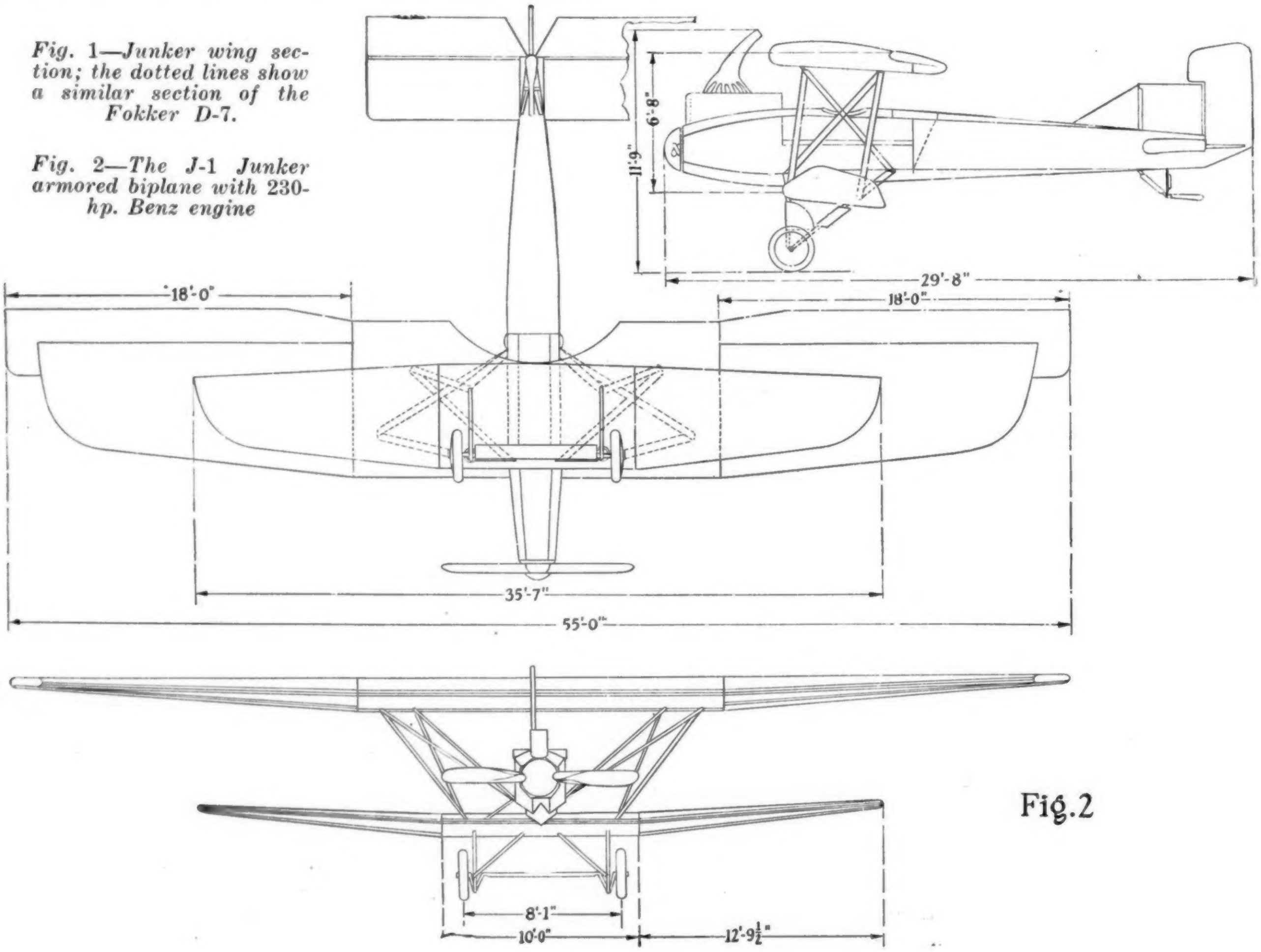
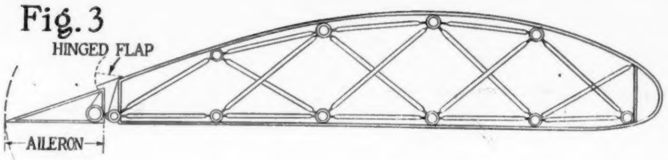


Fig. 1—Junker wing section; the dotted lines show a similar section of the Fokker D-7.

Fig. 2—The J-1 Junker armored biplane with 230-hp. Benz engine

Fig. 2

Fig. 3



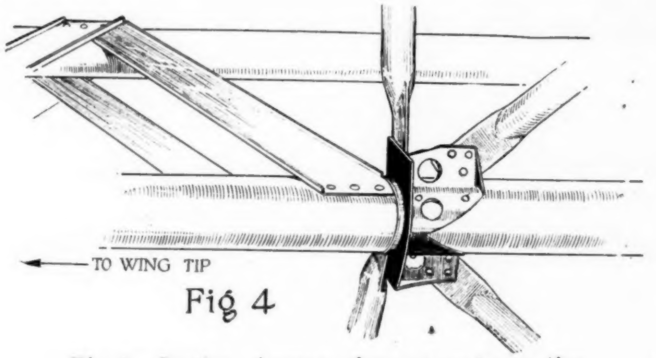


Fig. 3—Section of upper plane at center section
Fig. 4—Internal wing structure types

as far as possible, are gathered within the armored portion. The only wire employed is that in the rudder and elevator controls. No information is available concerning performance, but it is known that the machine requires an unusually long run in taking off.

The weights as painted on the fuselage are:

Empty	3724 lb.
Useful load	845 lb.
Total	4569 lb.

Other specifications are:

Engine	230 hp. Benz
Crew	Pilot, observer and gunner
Fuel capacity	26 gal.
Oil capacity	10 gal.
Power loading	19.9 lb. per hp.
Wing loading	8.56 lb. per sq. ft.
Total wing area	533.5 sq. ft.
Area of each aileron	32 sq. ft.
Area of rudder	15.4 sq. ft.
Area of fin	12 sq. ft.
Area of horizontal stabilizer	49 sq. ft.
Area of elevators	33.6 sq. ft

The machine is an armored biplane with cantilever wings and entire metal construction. It is intended for general service work. The ailerons and rudder are balanced, as in practically all German machines. There are two center sections with the fuselage supported between them. These center sections, with the fuselage and landing gear, form a unit to which the wings are attached.

The upper center section is much larger than the lower and is braced to the fuselage by a system of strong steel struts. The lower center section is built up in one unit with the landing gear. There are three groups of struts; one set from the upper to lower center sections, another from the upper center section to the fuselage, and the third from the lower center section to the fuselage. The pair connecting the upper center section and the fuselage form a cross and are prevented from touching by having their fittings at the upper plane—which they have in common with the interplane set—on different chord lines. These struts are of steel tube with aluminum fairing. They are joined to the spars by riveted steel collars carrying welded-on lugs. At the fuselage, they finish in fork-ends and are bolted to lugs welded to small steel plates riveted to the armor plate.

There are apparently two attachments which directly couple the fuselage to the center section unit. These are lugs fitted to the bottom edges of the octagonal body, midway between the forward pair of strut attachments. The lugs are bolted to corresponding lugs welded on to steel sleeves, which are in turn riveted to two upper spars of the lower center section. The aluminum cowl which bridges the gap between the body and lower center section is simply a fairing. As the machines examined were very much damaged the angles are hard to estimate, but it is believed that the angle of incidence of both planes is 3 deg. and the dihedral of the lower plane 3 deg.

The wings form the most interesting portion of the machine and do not at all follow common practice. There are ten tubular spars in the upper plane and six in the lower. With the exception of the front and rear spars, they are in pairs, one in the upper surface over one in the lower surface. They are braced to each other in such a manner as to form a warren truss between any two spars. Near the center section this bracing is tubing, and near the tips it is by means of strips longitudinally grooved. The bracing is fastened to the spars by steel collars on the spars to which they are riveted. The wings are joined to the center sections by threaded unions. Each spar is fitted with a steel sleeve which fits inside the duralumin tube and is riveted in place. One sleeve carries a threaded beveled collar and the opposite spar has a similar internal liner of steel, riveted in place, and a loose internally threaded steel collar. The end of the liner is beveled to take the bevel of the opposite spar. When fitted together, this makes a very rigid joint, and, when it is remembered that each spar has such a joint, it is evident that the wings are strongly attached to the center sections. The wing covering is corrugated duralumin sheet, with corrugations along the line of flight. The pitch of the corrugations is 1¾ in. and the depth ⅓ in. The thickness of the sheet is 0.015 in. The micro-structure shows that the material is cold rolled. Analysis shows the material to be "Duralumin." The weight is 3.65 oz. per sq. ft. Physical tests show a strength of 650 lb. per sq. in.

The ailerons are simple and consist of a steel tube at their lower leading edge, to which is riveted the lower surface. The upper surface is held by formers. They extend from the center section to the tip with the balancing portion beyond.

The fuselage is built in two parts, an armored box in front and a fabric covered duralumin tube structure aft. The armored part is made of 5 mm. (0.2 in.) steel plate. This armor is not added to an ordinary structure but com-

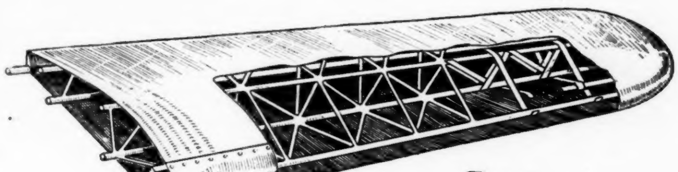


Fig.5

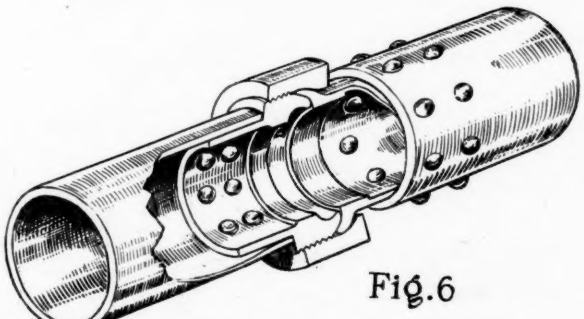


Fig.6

Fig. 5—Junker wing structure

Fig. 6—Spar joint

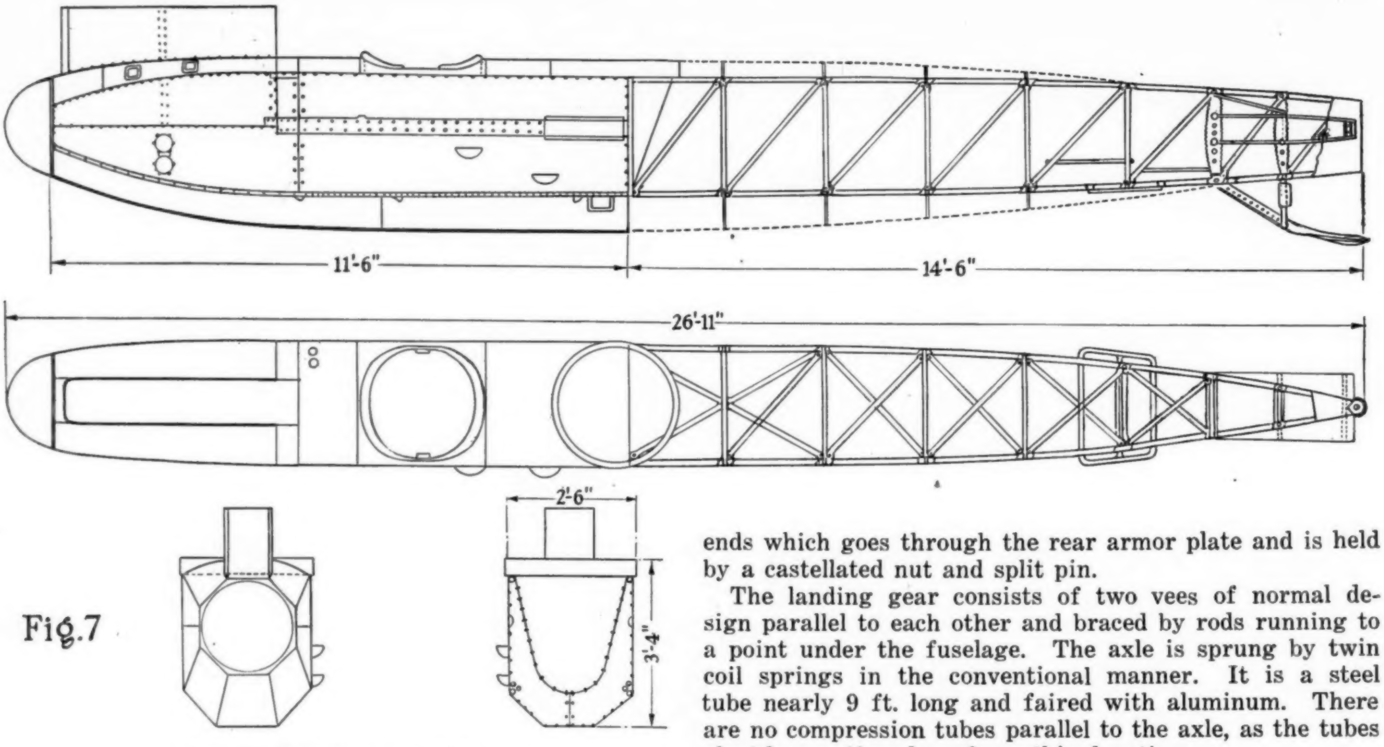


Fig.7

Fig. 7—Fuselage assembly

prises the fuselage itself. The armored unit houses the engine, fuel tank, pilot and gunner. The vertical cowl surrounding the engine is of armor plate and not a mere fairing. In the rear portion there are four longerons terminating in a vertical knife edge about 16 in. long. Diagonal duralumin tubes are used for internal bracing. The method is peculiar and worthy of careful attention. The junction of the duralumin tubes is effected by means of steel sleeves which embrace the longerons tightly and are pinned to them. The cross and diagonal tubes are flattened at their ends and riveted to shelves welded on to the clip. Three-ply formers are fixed to the upper and lower cross tubes and are joined by light wood stringers which pass from the end of the armored portion to the front of the tail skid.

The rear portion of the body is a structure of great strength. The vertical tubes in the last two bays before the stern post are replaced by strong bulkheads of duralumin sheet The longerons are not of the same diameter throughout but are spliced toward the rear of the fourth bay from the armored part. The splice is similar to those at the wing junction. The junction of the two parts of the fuselage is simple. The longerons have a stud at their ends which goes through the rear armor plate and is held by a castellated nut and split pin.

The landing gear consists of two vees of normal design parallel to each other and braced by rods running to a point under the fuselage. The axle is sprung by twin coil springs in the conventional manner. It is a steel tube nearly 9 ft. long and faired with aluminum. There are no compression tubes parallel to the axle, as the tubes already mentioned perform this function.

The control is by stick and rudder bar. The ailerons are operated by rods and levers. There is a rocking shaft in the wings which transmits the motion to the aileron tube. The rudder and elevators are operated through cables in the usual way.

The section, as well as the construction, of the horizontal stabilizer is approximately the same as the wing section. There are seven tubular spars, one in front and three pairs at intervals behind. The greatest depth is $7\frac{1}{4}$ in. The elevators have six hinges, three on each side. The construction is very simple, the covering being riveted to the elevator spar at the trailing edge. The rudder consists of a tube, a U-shaped leading edge of plain duralumin sheet and two halves of corrugated duralumin riveted together at the rear edge.

The tail skid is a substantial piece of ash bound with fabric, pivoted about its middle point, and fitted with a welded steel shoe. The shock absorber consists of the usual steel coil spring.

The fuel tank is an elaborate brass structure forming the pilot's seat. Benz pressure fuel supply is employed.

The radiator is under the upper plane over the rear part of the engine. Cooling effect is controlled by flaps which are set on the ground—a very elementary arrangement.

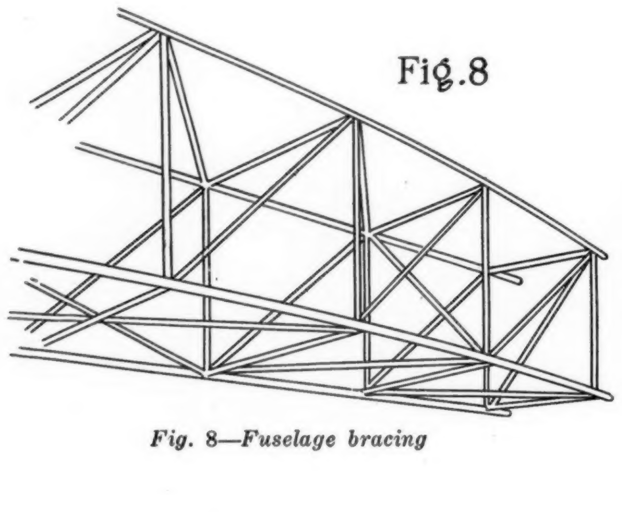


Fig.8

Fig. 8—Fuselage bracing

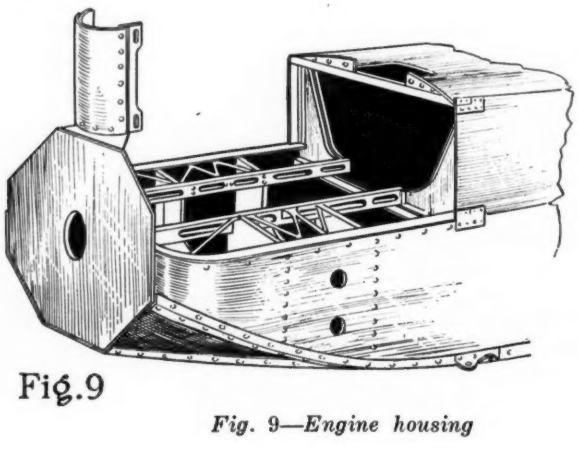


Fig.9

Fig. 9—Engine housing

A Six-Wheeled Truck with Tandem Rear Axle Drive

The growing use of pneumtaic tires on trucks introduces many new problems in truck design. A solution of some of them is given here in the six-wheeled truck. Instead of the usual rear construction, four wheels and two axles are employed with tandem drive. Uniform size tires are used.

By E. W. Templin*

THE development of the pneumatic tire for motor trucks has had an important influence upon the design of the trucks themselves. One of the problems introduced was how to equip heavy tonnage trucks with pneumatics. The Goodyear company has produced successful pneumatic truck tires up to and including 12 in. sizes, the largest size being suitable for the rear of 5-ton trucks. The fact that this tire is quite heavy and unwieldy brought about the consideration of the application of four small size, instead of two large tires in the rear.

Then, again, it was recognized that, while it is possible to build successfully pneumatic tires up to 20 in. diameter, it would be useless to build them above 12 in. diameter if that was the limit permitting of convenient handling in service. The service for which pneumatics could be used, then, would be limited to 5-ton capacity trucks. Adapting four tires to the rear of the motor truck then opens up the possibility of expansion from 5-ton capacity to at least 10-ton capacity.

Dual tires, while successful in a few cases and classes of service, and up to 5 in. in size, should not be considered for uses on $3\frac{1}{2}$, 5, 7 and 10-ton capacity trucks for the following reasons:

1. Inequalities of the road always throw an undue proportion of the load on one or the other tire; therefore, one of them would be carrying more than 50 per cent of the load.
2. In case of puncture all the strain is thrown onto one tire and the car may be driven for some time without the driver being aware of the puncture, causing the destruction of both tires.
3. In running over rough and stony roads, the chances of picking up rocks and thereby getting bad cuts is much greater than with the single tire.
4. In the case of puncture of the inside tire, it is necessary to remove both tires.
5. While it is possible to connect the valve stems of the two tires in order to equalize the air pressure, there would remain the objection that when one of them becomes punctured both would go flat, making it the more difficult to locate the trouble. Also, when striking obstructions the flow of air from one tire to the other would be too slow to make satisfactory equalization.
6. Difficulty of getting satisfactory rim equipment.
7. Difficulty of extreme width which puts severe stresses on the axle shaft.
8. Interference of inner valve stem with brake drum.

The other alternative is the application of wheels in tandem to the rear of the truck. Goodyear's first attempt at an application of this kind consisted mainly of an ordinary worm drive rear axle equipped with walking beams at each end, upon which were mounted the four rear wheels, each driven by chain from the main axle.

We were, however, inconvenienced by the chains jumping off and were not able to get a brake mechanism that would work. The main point against the design was its enormous weight. But it served to show us that satisfactory tire mileage could be secured from such an arrangement and that there was a good possibility of adapting four small tires in tandem to the rear of a motor truck.

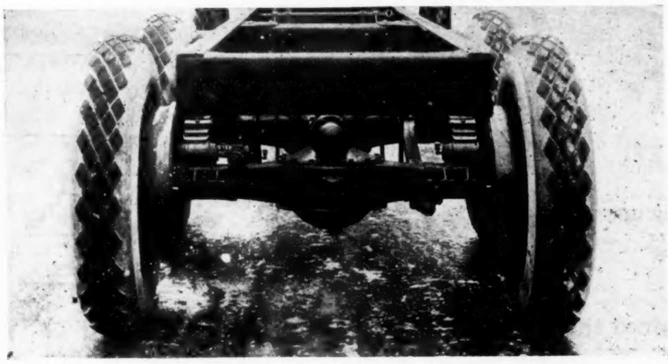


Fig. 1—Tandem axle construction of Goodyear six-wheeled truck

In order to further develop this point, we built up the tandem axle construction shown in Fig. 1. This construction appears to have good possibilities and has at present operated some 10,000 miles, 2000 of which were on very severe and rough roads under full load.

This assembly was adapted to a standard make of truck which was somewhat short of engine power and transmission gears to give satisfactory road speeds and low gear ability. It was, therefore, decided to design a new and complete truck of 5 tons capacity specially suited to pneumatic tire equipment and capable of long continuous runs at comparatively high speed. A side view of this truck is shown in Fig. 2.

Basis of Design

The following points were laid down on which to base the design: Estimated total weight of truck, body and load, 21,000 lb.; normal speed, 25 m.p.h.; desirable to ascend 2 per cent grade on high gear at full speed; low gear ability sufficient to slip rear tires on dry concrete under normal load; cost of manufacture to be close to cost of standard 5-ton solid tire truck. Truck design to be conventional in respect to weight distribution; body 16 ft. long.

*Motor truck engineer, Goodyear Tire and Rubber Co.

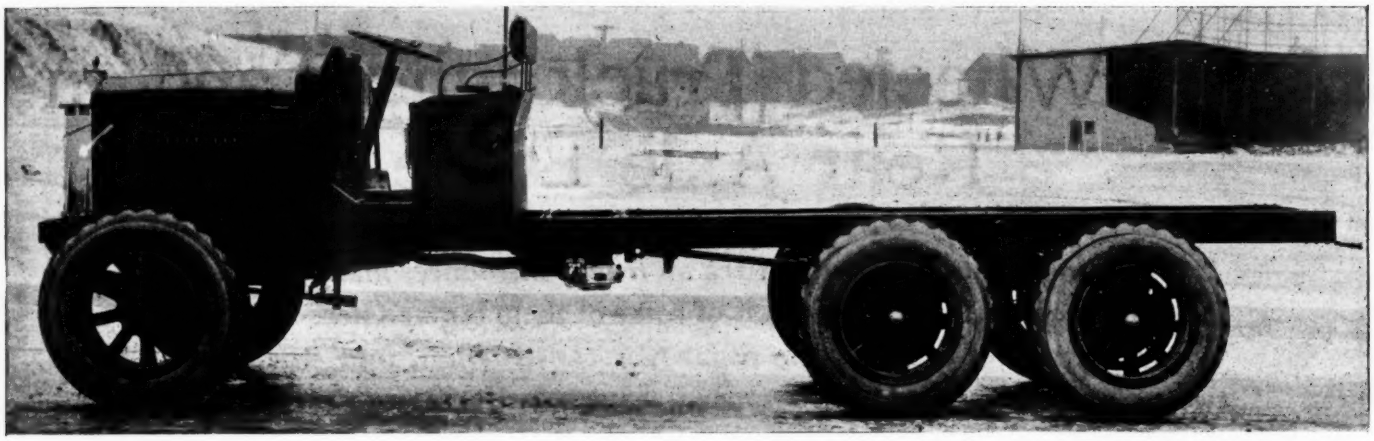


Fig. 2—Side view of Goodyear six-wheeled truck

Estimated weight on rear tires, 76 per cent of total, or 16,000 lb.; on front tires, 24 per cent of total, or 5250 lb.

Tire Size	Extreme Maximum Allowable Load Per Tire	Inflation Pressure
34 x 5	1700	80
36 x 6	2200	90
38 x 7	3000	100
40 x 8	4000	110
42 x 9	5000	120
44 x 10	6000	130
48 x 12	8500*	140

*Not standard yet with S. A. E.

It will be seen from the table that the 40 x 8 in. tire is proper for the rear, whereas the 38 x 7 in. tire would be sufficient for the front. However, it was decided to use tires of the same size in front and rear to gain the advantage of uniformity.

A normal speed of 25 m.p.h. requires the rear tires to rotate at the rate of 210.1 r.p.m. In consideration of the fact that 1200 ft. p.m. piston speed is a reasonable rate for present day gasoline engines and that a 6-in. stroke engine is required, the gear ratio in high becomes

$$\frac{1200}{210.1} = 5.71$$

A ratio of 5.8 was determined upon as being the closest practical ratio.

Tractive Factor

By observation it was determined that the truck should be capable of ascending a 2 per cent grade at full speed (25 m.p.h.) on smooth, hard road. The various components of the resistance encountered under these conditions per pound of truck and load are as follows:

Wind resistance	0.01 lb.
Road resistance	0.01 lb.
Grade resistance	0.02 lb.
Total tractive factor	0.04 lb.

The low gear tractive factor should be 0.50, or sufficient to slip the rear tires in case of necessity.

Engine Torque Required

Tractive factor (high gear) = 0.04 =

$$\frac{\text{Engine torque} \times \text{high gear reductions} \times \text{transmission efficiency}}{\text{Weight of truck and load} \times \text{radius of rear tires}}$$

$$0.04 = \frac{\text{Engine torque} \times 5.8 \times 0.90}{21000 \times 20}$$

$$\text{Engine torque} = \frac{21000 \times 20 \times 0.04}{5.8 \times 0.90} = \frac{16800}{5.22}$$

$$= 3220 \text{ lb.-in.}$$

It is important to note that this torque is to be sustained at 1200 r.p.m. of the engine.

An analysis of engines on the market indicated that the Hercules Model "T3" four-cylinder engine of 5-in. bore and 6-in. stroke would develop satisfactory torque, and it was therefore determined upon. The performance of this truck in the 2000 to 3000 miles of preliminary operation indicates that the engine combined with the gear ratio is indeed satisfactory.

Transmission Low Gear Reduction

Tractive factor (in low gear) = 0.50 =

$$\frac{\text{Eng. torque} \times \text{total low gear reduct.} \times \text{eff. of trans. in low}}{\text{Total weight of truck and load} \times \text{radius of rear tires}}$$

$$0.50 = \frac{3200 \times \text{total low gear reduct.} \times 0.85}{21000 \times 20}$$

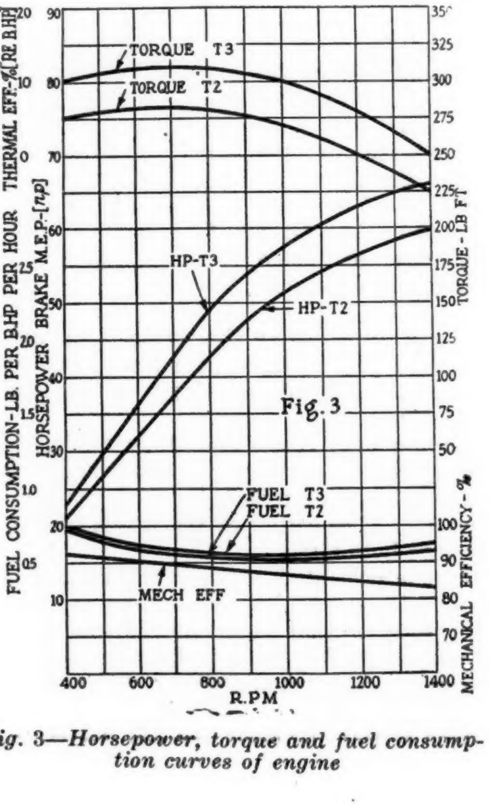


Fig. 3—Horsepower, torque and fuel consumption curves of engine

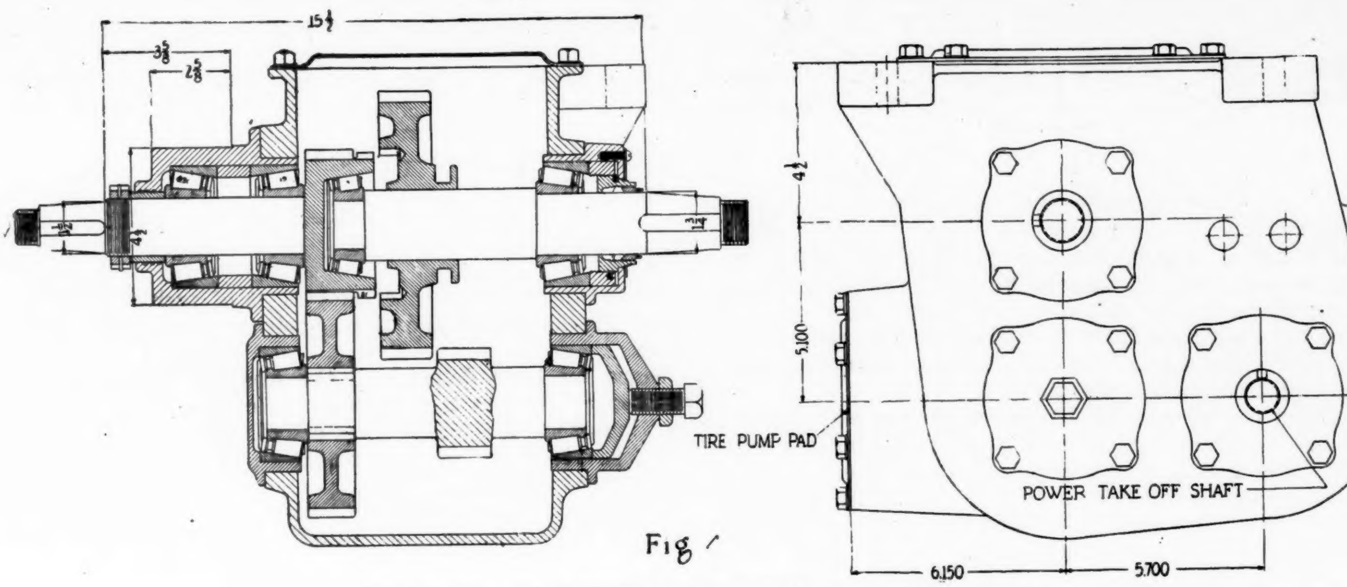


Fig. 4—Two speed auxiliary transmission on Goodyear truck

$$\text{Total low gear reduct.} = \frac{21000 \times 20 \times 0.50}{32000 \times 0.85} = \frac{210000}{2720}$$
$$= 77.2$$

With an axle reduction of 5.8 to 1 the low transmission reduction becomes 77.2/5.8 or 13.3; a reduction of 14 to 1 is used. This reduction is difficult to obtain in a conventional type of transmission, so it was decided in conference with transmission builders to use a three-speed unit power plant transmission on the engine and a two-speed auxiliary transmission placed amidships.

Specifications

Carrying capacity, 5 tons; wheelbase, 180 in. (measured from center of front wheel to center between rear wheels); tires, 40 x 8 in. pneumatic all around.

Engine: Hercules Model T3, 5-in. bore, 6-in. stroke, 4 cylinders; location, under hood; crankshaft bearings, 5; horsepower rating, 40; torque, 3200 in.-lb. at 1200 r.p.m.; cylinders cast in pairs; valves placed on left; water circulation by pump; radiator, Modine "Spirex"; ignition, Eisemann magneto; governor, none; carbureter, Stromberg; lubrication, force feed; motor suspension, 3 point; motor weight, 1016 lb.; valve diameter, 2⅛ in.; crankshaft diameter, 2⅜ in.; crankshaft bearings (Nos. 1 to 5), 2⅜ x 2¼ in., 2⅜ x 2¼ in., 2⅜ x 2¼ in., 2⅜ x 2¼ in. and 2⅜ x 4 in.; camshaft bearings (front, center and rear): 1¼ x 3 in., 1¼ x 2¼ in., 1¼ x 1¾ in.; connecting rod bearings, 2⅜ x 3 in., 2⅛ x 1⅜ in.

Transmission: Brown-Lipe Model 60 U.P.P. and auxiliary; Brown-Lipe clutch Model 60, multiple disk type; six speeds forward and two reverse. Gear ratios: Low, 82; second, 35.7; third, 23.2; fourth, 20.1; fifth, 10; sixth, 5.8; low reverse, 99.5; high reverse, 28.4.

Goodyear tandem axle drive, using two Stanpar Model 603, worm drive rear axles equipped with 21-in. diameter brakes. Ratio, 5.8 to 1. Torque absorbed in tandem axle assembly. Propulsion is taken by the springs to the frame. There are four brake drums on the rear wheel of each drum, 21 in. in diameter and equipped with two sets of shoes each 2¾ in. wide, which makes a total width of 5½ in.

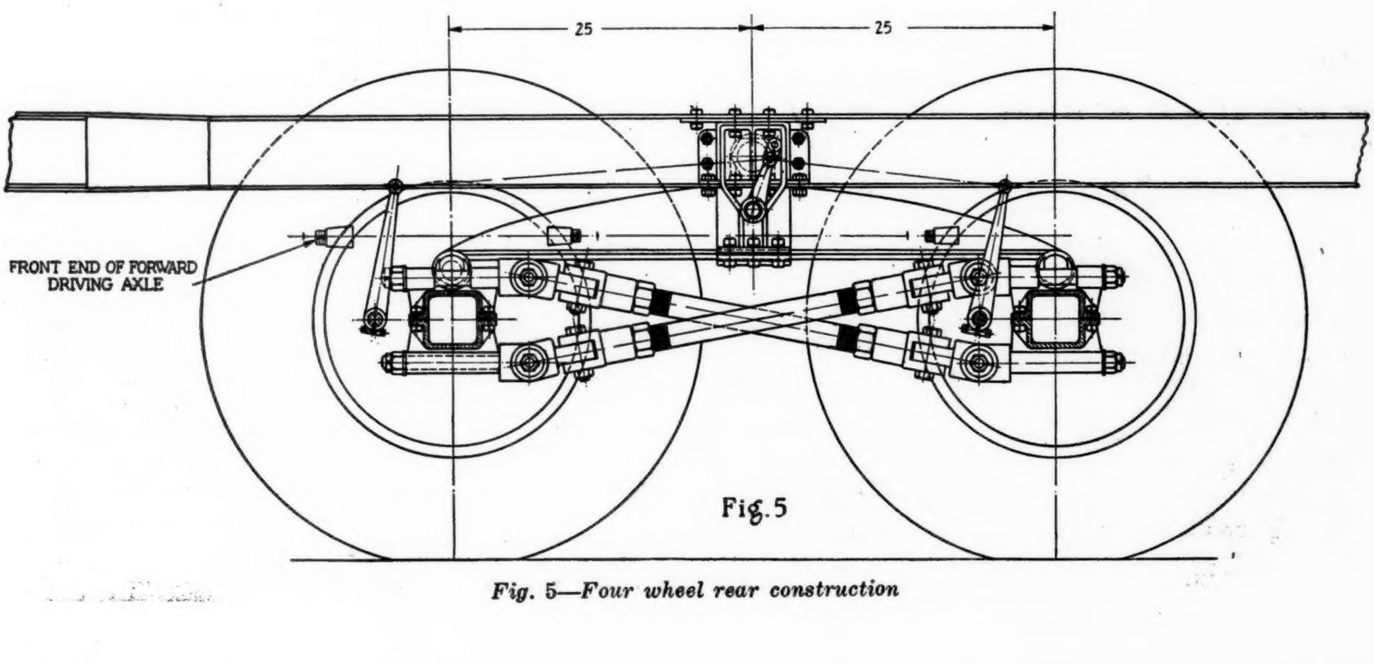


Fig. 5—Four wheel rear construction

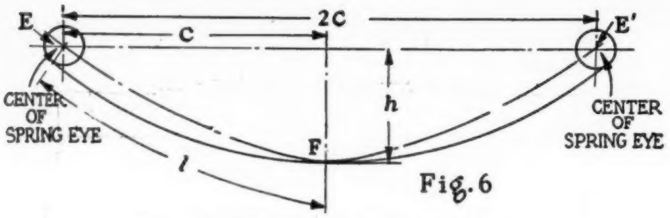


Fig. 6—Diagram of chassis spring

The design of mechanism for operating the brakes on two axles connected in tandem, by the Goodyear method, contemplates a layout of brake levers and rods in such a way as to compensate for (1) rotation of the tandem axle assembly about its axis, (2) deflection of the springs and (3) a slight rotation of the axles.

The first requirement is that the upper end of the operating rods be pivoted on or very near to the common axis of the tandem axle assemblage and then determine the position of the brake lever eyes so as to accomplish full compensation. The method of determining the positions for the brake lever eyes is as follows:

As the camber of the spring and the distance between spring eyes in the loaded condition are predetermined values, the relative positions of the spring eyes in the light and extreme loaded conditions remain to be found. In this case the spring in a loaded condition has a camber of 3/4 in. and a length between spring eyes of 50 in.

Referring to Fig. 4, assume that points E E' lie on a parabola whose vertex is at the center of the top of the main leaf of the spring at F. (This assumption is accurate within about 1/16 in., as determined by trials.)

Let L = length of parabola between E and F
C = one-half the distance between the eyes of E E'
H = camber of spring

Then

$$L^2 = C^2 - \frac{4}{3} H^2 \ldots\ldots\ldots\ldots\ldots (1)$$

Knowing the values of C and H in the loaded condition from the specifications of the spring, this equation may be solved for L. Assuming that the value of L remains constant for any deflection of the spring, the value of C may be determined for any value of H.

$$C^2 = L^2 - \frac{4}{3} H^2 \ldots\ldots\ldots\ldots\ldots (2)$$

$$C = \sqrt{L^2 - \frac{4}{3} H^2}$$

Applying equation (1) to the spring in the loaded condition, for which $2C = 50$ in., $C = 25$ in. and $H = 0.75$ in.,

$$L^2 = C^2 + \frac{4}{3} H^2$$

$$L^2 = 25^2 + \frac{4}{3} \times 0.75^2$$

$$L^2 = 625 + 0.75 = 625.75,$$

a constant to be used in the following cases. For the extreme loaded condition $H = 0.25$ (reverse camber) $L^2 = 625.75$. Applying equation (2)

$$C = \sqrt{625.75 - \frac{4}{3}\ (0.25)^2}$$

$$= 25.082 \text{ in.}$$

$$2\,C = 50.164 \text{ in.}$$

The total camber when under light load is 3¾ in. Since

$$C = \sqrt{L^2 - \frac{4}{3} H^2}$$

$$C = \sqrt{625.75 - \frac{4}{3}\ (3.75)^2}$$

$$C = \sqrt{625.75 - 18.75} = 24.63$$

$$2\,C = 49.26$$

Therefore the amount of movement of each spring eye from the loaded to the light position is approximately 3/8 in.

Since we have determined the location of the spring eye centers in the three conditions of load, the relative position of the brake lever eyes of both driving axles, with reference to their vertical center lines, may be readily found and laid out.

Having now located the brake lever eyes of the forward driving axle in three positions, connect the loaded and light points with a line and erect a perpendicular bisector to same, thereby finding a point of intersection on the vertical center line of the spring, which is also the vertical center line of the tandem axles. Since this is the common center of an arc passing through these two points, the first point will fall upon or approximately upon the same arc.

We have now found the center point of the spring trunnion or the pivot point of the axle unit, while the length of the brake levers of the rear driving axle remains still to be found.

We have previously found the relative position of the brake lever eyes on the forward driving axle, with reference to the vertical center line under three conditions of load. Now draw lines, the same relative distance from the vertical center line and parallel to it for the same three conditions of load. Now, again using the common pivot point of the axle unit as a center, draw an arc intersecting these three lines. The proper points of intersection determine the length of brake levers on the rear driving axles and insure a brake mechanism that will compensate for any deflection or spring or oscillation about the pivot point.

Chassis Specifications

Frame, pressed steel, 7 in. deep, 3 in. flange, ¼ in. stock; maximum stress under normal load 10,000 lb. p. sq. in. Springs, front, 3 in. wide, 46 in. long; rear, 4 in. wide, 50 in. long. Steering, left hand. Control,

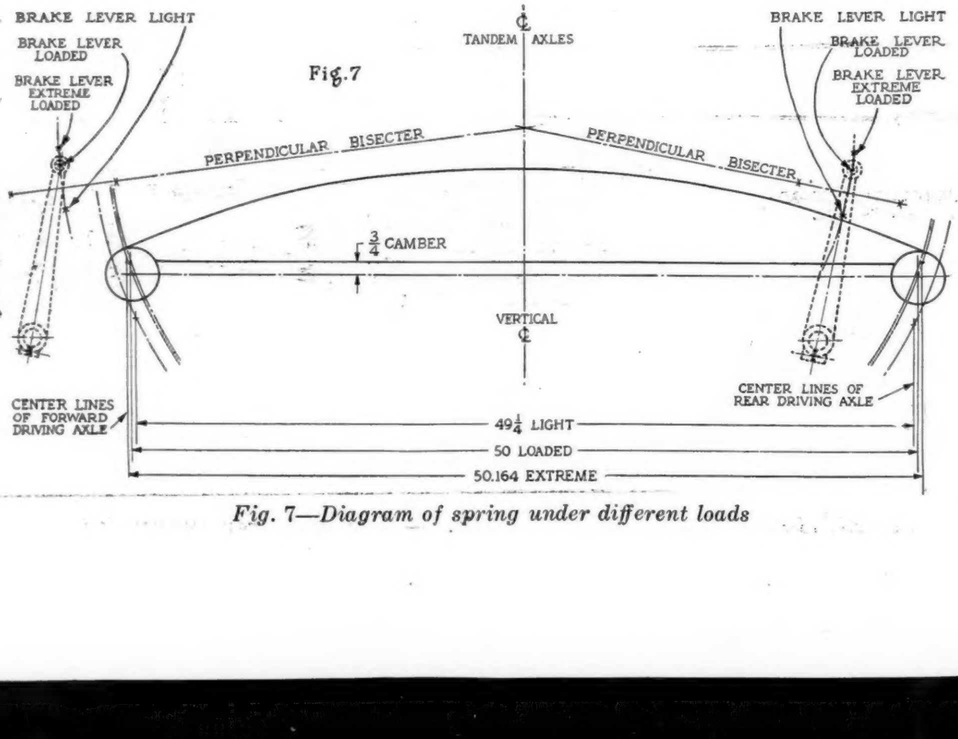


Fig. 7—Diagram of spring under different loads

center. Front axle, Standard Parts Co. Model 507. Tire pump, Kellogg Model 42 mounted on transmission. Chassis weight, total, 8520 lb.; front, 3810 lb.; rear, 4710 lb. Tread, front, 66 in.; rear, 61¼ in. Total weight (loaded), front, 4520 lb.; rear, 16,500 lb. Fuel tank capacity, 50 gal. under seat, 17 gal. dash feed tank. Oil tank capacity, 10 gal. under seat. Loading space, 15 ft. Normal motor speed, 1200 r.p.m. Normal truck speed, 25 m.p.h. Turning radius, 35 ft.

After this truck was built up it was interesting to note whether or not it would perform as expected. Trial runs under full load and limited to 30 m.p.h. maximum speed showed the truck capable of sustaining an average of 21 m.p.h. on the roads between Akron, Canton and Cleveland. In one case the truck became "ditched," but it was able to get out under its own power. In another case the truck pulled out a 5-ton truck stuck at the side of the Cleveland road (under 8½ tons load). The Goodyear truck pulled until the rear wheels slipped on dry brick pavement. In one minute after making this heavy pull the truck was on its way at 25 m.p.h.

On a trip to Boston over the National Highway the truck averaged 10 m.p.h. in the Cumberland Mountains, ascending the mountains at 6 m.p.h. fully loaded.

Stability of Motor Omnibuses

IN London, where the motor bus is the principal means of transportation, there have recently been a number of accidents due to the buses overturning, and the *Engineer* has investigated the conditions of stability of these vehicles.

Taking the ordinary design of motor omnibus used on the London streets, we find that it is a vehicle weighing, in service condition, some 9150 lb., of which weight a little over 64 per cent is in the chassis. It is capable of carrying, without overcrowding, thirty-six people, representing, on the average, a load of 5400 lb., so that when fully laden the vehicle and those on it weigh something in the neighborhood of 6½ tons. Its center of gravity, in service condition, but without load, lies low down, being only about 44½ in. above the road surface, or just a little above the level of the tops of the rear wheels. The gage of the wheels being 70 in., it follows that the vehicle will recover itself if it is tipped up until the rear axle is lying at 38 deg. to the horizontal.

When the omnibus is fully loaded the center of gravity rises to 62 in. above the road surface, so that in this condition the maximum heel from which the vehicle will recover itself is 29½ deg. These figures do not take account of the fact that the body is mounted flexibly on the chassis. In an actual test the yielding of the springs permitted the vehicle to be tipped, when empty, until the center line of the body was inclined at 46 deg. to the vertical. It will be seen, then, that the stability of a motor omnibus empty or fully loaded is very considerable when the vehicle is stationary.

When the omnibus is in motion the angles of heel mentioned above are a measure of its stability so long as its course is a straight line, and indicate the magnitude of the "bump" in the road required to upset the vehicle. Thus the wheels on one side of the omnibus could rise off the road surface nearly 3 ft. without entailing the upsetting of the vehicle. Bumps of this description can be ruled out of consideration.

The critical test for the stability of the vehicle arises, of course, when centrifugal force comes into play during the rounding of a corner. Taking 30 ft. as the radius of the curve and 7 m.p.h. as the speed when rounding the corner—both figures being on the severe side—the upsetting force on the fully loaded omnibus works out at 7915 lb./ft. The righting moment of the vehicle is 42,430 lb./ft. when on the straight, so that on rounding a corner of the stated radius at the stated speed the stability of the omnibus is reduced by a little over 18 per cent.

The Third Brush Regulation in Automobile Generators

(*Continued from page* 1301)

would give 110 amps. if the only loss were armature resistance. Brush loss brings the current down to 75 amps.; reactance to 55 amps.; field distortion to 32 amps.; and then changing the excitation from shunt to third brush reduces it to the final rate, 14 amps.

In ordinary non-interpole generators, commutation is improved by shifting the brushes forward under the tip of the leading pole, thereby bringing the coils undergoing commutation into an assisting field. But in the third brush machine under load condition, shifting either forward or back of neutral brings the coils into an unfavorable field. In Fig. 12 it would be necessary to shift to *B'* to reach a favorable field. Fig. 13 shows the practical effect of changing the brush contact on a standard generator having ¼-in. brushes. It shows why the output of this class of machine may change 50 per cent after being installed on the car in case the brushes are not well ground in.

From the theory that has been given the following operating points are deduced:

1. Poor connection in the charging line increases the amperes output.
2. Increasing the main brush or third brush width, decreases the output.
3. Setting the third brush to give higher output than normal is dangerous because the trailing edges of the main brushes will be burned away, resulting in still higher output.
4. Setting the main brushes ahead of the manufacturer's position is dangerous because a larger field current will be drawn for a given output.
5. Setting the main brushes back of neutral is dangerous because it will cause heavy short circuit currents to flow through the main brushes, eating away the trailing edge.
6. Setting the brushes on neutral by finding the position of "no rotation" with only armature current flowing, is correct only with generators having symmetrical pole faces; that is, equal airgaps at the entering and trailing edges.

Rigid and Semi-Rigid vs. Flexible Mountings in Creeper Tractors

The rigid type of mounting meets all the conditions of ordinary operation, is much easier on the track chains, and is preferable to any other type for use on the creeper tractor. After summing up the purposes of track-laying construction the writer proves this thesis in the following discussion.

By E. F. Norelius

THE development of vehicles of the track-laying variety extends back for a good many years, and it is quite necessary to have knowledge of this development to appreciate fully the problems that were encountered. Like problems were met with in other lines. The development of the railroad locomotive has taken years. The first type had only one pair of drive wheels. Then followed the American, Atlantic and Pacific types for fast passenger service, where high speed was essential, and the Mogul and finally the consolidation and decapod types for heavy freight service. Tractor development compares favorably with that of the railroad locomotive.

One of the laws of physics is that the coefficient of friction is a fixed quantity irrespective of the load applied. This is not entirely true, but is practically true within the supporting limit of the material. High tractive efforts were desired in the freight division of the railroads, and to accomplish this it was necessary to use a great many drivers, as there are no materials that would stand the heavy concentration of load that would result by placing the locomotive weight on one or two pairs of drivers.

Consider then the problem presented where the surface over which the equipment has to travel is the ground, and a vehicle with numerous drivers is to be designed so that it can make short turns, as required in tractor work. The modern railroad locomotive would be impossible if it had to make short turns. Of course, a tractor with four driving wheels, with independent steering at the front, at the rear, or both, is not an impossibility and may be a development of the future, similar types having been successfully developed in the truck field. But to make more than four wheels drivers would be next to an impossibility.

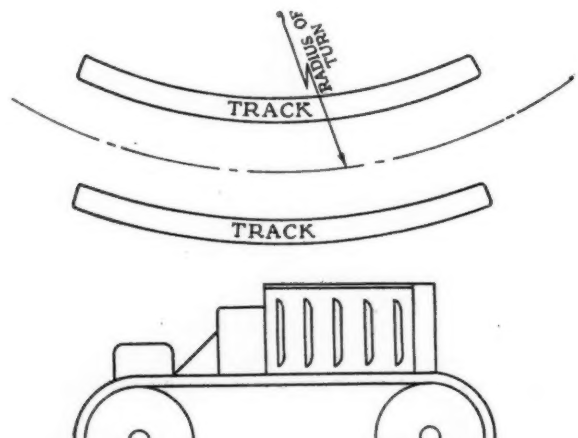


Fig. 1

The early inventors, when approaching the problem of steering, did not consider it advisable to slide the track around, as is done at present, but thought it necessary to have the track flexible in all directions like a rope, so that it could be bowed to the proper radius to make the turn, as shown in Fig. 1. This, of course, led to complication and a track construction that it was next to impossible to make durable. This idea was never commercialized. The present method of slipping tracks around appears crude and clumsy, and it actually is, but it leads to simplicity and cheapness of design.

The question suggested by the heading of this article is whether it is best to have a more or less rigid truck or one which is flexible to the extent that it will make the track conform to all irregularities of the ground surface, like the Renault type. Before going directly into a comparison of these different types (Figs. 2, 3 and 4) it may be well to visualize the purposes of the track-laying construction. These purposes may be summed up as follows:

1. To distribute the weight of the machine over a greater ground surface and thus reduce ground depression, so as to be better able to operate over soft ground.
2. To distribute the tractive effort over a greater surface and thus have the shear value of a greater area of soil.
3. To bridge uneven places in an endeavor to have smoother riding qualities and not continuous bumping as when the tractor drops into every depression.
4. To have a more compact unit with low center of mass and narrow width, making it more suitable for all classes of operation.
5. To have a unit capable of more flexible control, making short turns and giving quick action in tight places.
6. To increase the efficiency of transmission from powerplant to drawbar.

By a study of Figs. 2 and 3, it can be seen that this type comes nearest to accomplishing purposes Nos. 3 and 6 and that the type shown in Fig. 4 most nearly achieves purposes Nos. 1 and 2. But it is worth while here to consider this analysis more in detail to see if these conclusions are correct.

Considering purpose No. 1, would it be better, when the tractor is operating over soft ground, to have the

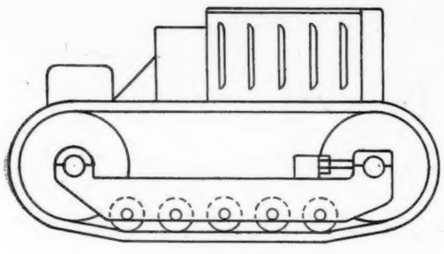


Fig. 2

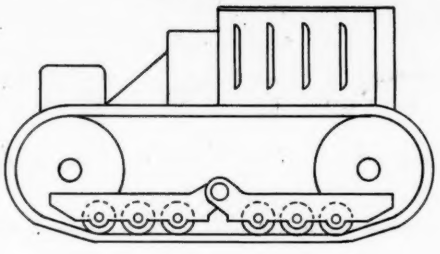


Fig. 3

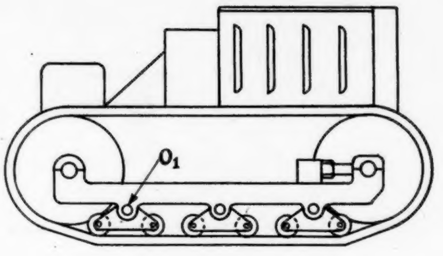


Fig. 4

track chain conform to the nature of the ground, as would be the case with the construction shown in Fig. 4, or would it be better to have the more rigid type of construction where the pressure on the high points would depress them to the level of the low points? Is the large percentage of land and roads over which the tractor operates of such a character as to present a succession of small hummocks and hollows, or are the ground undulations of a more gradual nature? It is quite evident that operating conditions do not call for extreme flexibility and that for operation on soft land a more rigid construction meets the requirements just as well.

In connection with purpose No. 2 it is generally agreed that an ordinary wheel tractor answers all requirements under conditions of level land, hard, firm surfaces, no hollows of material size and no soft places through which it must operate. If a wheel tractor is satisfactory under these conditions, then it must be evident that a tractor which bridges the hollows and which has sufficient ground contact area to travel over the soft places answers all requirements of service, and it is not necessary to fit into the depressions in order to get the extra shear value of the soil.

There can be no argument on purpose No. 3, as a close study of the diagrams shows clearly the relative merits in this regard. Purposes Nos. 4 and 5 are accomplished equally by both devices.

Purpose No. 6 has been mentioned above as being best accomplished by the more rigid type. It will certainly be admitted that it takes energy to bend a chain as it travels under load. This loss of energy decreases the efficiency and causes wear at the pin joints. Further, rail wear and the stress upon the rollers must be worse on the more flexible or Renault type, as the rollers do not approach the succeeding link tangentially but strike it at an angle depending upon the nature of the ground surface.

The object aimed at in the design of the Renault type, no doubt, was to get a more uniform distribution of load on the track wheels, and perhaps for ordinary travel it does accomplish this purpose. But the best that can be done is to distribute the load over the two wheels, and it is recognized that any bearing which is properly designed will stand twice its normal load for an instant, and that it is the average load that is of real importance.

Figs. 3 and 4 bring out another feature in respect to which the Renault type is at a disadvantage, particularly in the case of short tractors with few truck wheels. The point of overturning has been moved forward to O_1 on the Renault type, due to the equalizer between the two rear rollers, and the stability therefore has been reduced. The tractor, when pulling a load, has a much greater tendency to ride low at the rear and high at the front.

Fig. 4 brings out another feature which is a very serious argument against any truck of less than three truck wheels. In the Renault type there must necessarily be trucks of two rollers each. If these are operated over a jointed track on soft land there is no way of arranging them to give smooth operation. If the distance between the two rollers which are equalized, is a multiple of the pitch of the track, then both wheels will strike joints at the same time and cause depression of these joints as shown in Fig. 5A. The wheels must now rise out of these depressions to get back to the center of the link, as the weight on the wheels will not depress the whole link as deeply as they will the ends only. If, however, the center of the rollers is as shown in Fig. 5B, then each roller will alternately cause a depression at the joint and a quick, small oscillation of the truck, one position of it being shown in Fig. 5B. This oscillation will cause a vibration which can easily be detected in the rest of the vehicle.

It would, therefore, appear that the rigid type meets all the conditions of ordinary operation and is preferable, and that it is much easier on the track chains, which is the big item to be considered in this type of equipment.

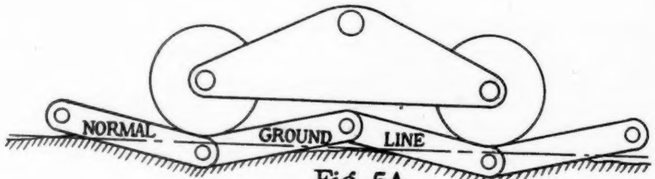


Fig. 5A

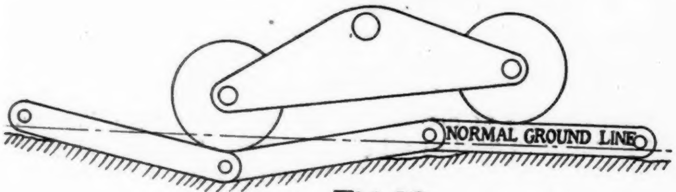


Fig. 5B

OWING to its extensive employment in the manufacture of iron and steel, manganese was in great demand during the war. Unfortunately there was a serious shortage of manganese ores, as supplies from the Caucasus, the chief producer, were cut off, and shipping difficulties restricted the amount available from India, which ranked second only to Russia as a producer of manganese ores. At the present time the output from India is increasing and Brazil has enormously enlarged its production; but in view of the disturbed conditions still prevailing in Russia there is likely to be a continued shortage of the ore there for some time to come, particularly of the higher grades now required by metallurgists and in chemical industries. Several additional sources are indicated by the Mineral Resources Committee of the British Imperial Institute as possible contributors to the world's supply in the future.

The FORUM

A Statement Regarding Steam Automotive Apparatus

Editor AUTOMOTIVE INDUSTRIES:

THERE are certain fundamental reasons why the automotive engineer should pause in his intensive efforts to further improve the gasoline power plant, to impartially consider whether he is pursuing the line of least resistance. It is the function of the automotive engineer—as of all other engineers—to not only apply the achievements of the arts and sciences to the use of mankind, but to do so in such a way as to make the dollar go farthest. In other words, it is not a technical solution that he should aim for, but a commercial solution. This, of course, is the avowed attitude of engineers, but it is at least a question whether in the automotive field the horizon has not been limited to internal combustion engines. Steam, as a possible alternative, if thought of at all, has usually been passed up as unworthy of consideration—as having been tried and found wanting.

But if the automotive engineer preserves the broad mental outlook referred to above, he will not be satisfied to abandon a possible chance for advancement because somebody else did not make a success of it formerly and under other conditions, and moreover he will be quick to realize that a great change in fundamental conditions may turn what was formerly a failure into a present success.

My present purpose is simply to draw attention to these fundamental conditions, to indicate why in the past so little has been done with steam and to point out what should be, in my opinion, the logical line of development.

Fundamental Conditions.—These are primarily:

(1) The Fuel Situation.
(2) The Degree of Perfection Demanded.
(3) Method of Attaining It.

The fuel situation is too well known to need extended comment. The increasing price and decreasing quality of gasoline, of which we have by no means seen the end; demand increasing out of all proportion to production, are matters of common knowledge. It is absolutely essential that automotive apparatus (automobiles, buses, trucks and tractors) be able to effectively use a far larger proportion of crude oil than is represented by its gasoline, or even its kerosene content. The steam automotive power plant easily uses any liquid fuel.

A tremendous degree of money, energy and ability of the highest order has been and is being brought to bear aiming at using the lower grade fuels in internal combustion automotive apparatus. Remarkable research accomplishments have been achieved. The tenacity of purpose displayed might be likened to that of a bulldog who refuses to slacken his grip, even though he might make much greater headway by doing so, for the problem attacked is not "how to get power from the fuel," but "how to get it by means of the internal combustion engine."

A survey of the proposals now extant aiming at maintaining the internal combustion engine in its present predominating position, displays the stress of the situation. Diesel type engines and gasifying auxiliary plants are among these. Surely such proposals as these should make engineers give pause to see whether an automotive type steam plant is not a better commercial solution.

The other fundamental condition, the degree of perfection demanded, needs a moment's thought, more particularly because it has a distinct bearing on the question next to be considered—the past history of steam apparatus. But it is not necessary to go into details here. We all know what is demanded to-day in automotive power plants, even if we don't all equally realize what an awe-inspiring cosmos the modern automotive plant is, with all its frills and auxiliaries. Yes, awe-inspiring is the word, for divorce yourself for a moment from that "contempt" (acceptation) bred of familiarity, and think for a moment of the herculean task and boldness of undertaking to place into the hands of the totally unskilled, a multi-cylinder exceedingly high speed engine, depending on an ignition system having an array of small and delicate parts, and still more delicate adjustments, equipped with a variety of finely posed automatic devices for lubricating, tooling, controlling, etc., supplemented by an additional power plant consisting of electric motor, generator, battery, automatic switches and instruments—to be operated far out of reach of help, without knowledge for the most part of what the apparatus is for, what it does or how it does it—almost without knowledge that it exists—does not the contemplation of a plant which will meet this ordeal inspire awe? And when we are told that we are not yet through with its elaboration, that more parts and devices are necessary, does not that awe almost turn into dismay? We don't realize this because we are immersed so deeply in it, because we don't think of it all at once, but just a little piece at a time.

My purpose in making the above remarks is not to underrate the high perfection of the modern gas power plant—not to minimize its endurance and reliability or its almost complete attainment of ends sought—not to belittle the splendid ability of the engineers to whom credit is due for this perfection—my object is to invite engineers to consider, in true perspective, their goal and their route to that goal, with a view to obtaining an equally true perspective of the route via steam, so as to reach an impartial decision of their relative commercial merits—to ask with all deference whether they are not emulating the bulldog referred to above, in gripping internal combustion engine problems, instead of letting go and getting a grip on something better. But the first step to obtaining these true perspectives is to realize that in the case of gasoline apparatus their familiarity with its details causes engineers to tacitly accept or pass over its commercial abnormalities, while their unfamiliarity with the details of steam apparatus tends to magnify in their minds its difficulties.

Past History of Steam.—On this subject, also, it is not my purpose to deal at all exhaustively, but only to draw attention to the important facts which have influenced the commercialization of steam automotive apparatus.

Say "steam car" to the average automobile man, and he nearly always answers, "Oh, yes; White and Doble." The few better informed add, "Stanley." The first two names, however, mostly depict the conception of a 1920 steam car. The fact that the White car in mind was nearly contemporary with the Cadillac single cylinder ("one lunger"), and that it was therefore no more representative of a 1920 steam car than the one lunger is representative of a 1920 Cadillac, is not appreciated.

As for the Doble car, it was never in production or marketed. It was purely an experimental development, though great sums were spent on publicity. The interest aroused by this publicity was, however, significant.

Quite the opposite is the case of the Stanley. This has been both technically and commercially successful for a long period of years, notwithstanding a relatively limited output. The reasons for this limited output, however, were purely administrative and corporate and entirely dissociated from the technical and commercial merits of the car. The modern Stanley steamer is a superb car, having the grace and ease of movement of an electric, coupled with the power and speed characteristics of the speedster. It is unlike anything else in the pleasurable nature of its performance. Yet its development has depended on cut and try methods of so unorganized a character that it is certain any gas car depending for its development on a similar research department would have been a total failure. The inevitable deduction from this is that if anything like the degree of research and experimental work had been devoted to the perfecting of the steam car that the gas car has enjoyed, the former would have far outstripped the latter in its commercial advancement.

Logical Line of Development of Steam Cars.—Notwithstanding the foregoing remarks about the Stanley car, I venture to believe that the commercialization of steam cars will be more quickly achieved with a design differing in some important particulars from the Stanley car.

The Stanley, the Doble and other designs now being prepared for the market by other concerns that I have not mentioned are all in the high-priced class. Stanley prices are $3,700 to $5,600. Furthermore, the effort to attain high ideals has prevented the adoption of standardized parts to anything like the extent prevailing with gas cars. This has contributed to high prices and limited production.

The high price range has thrown steam cars into competition with the most highly developed and perfected gas cars whose performance leaves but little more to be desired. Consequently the still greater superiority of steam performance is regarded as superfluous by those who have not had opportunity to know the difference. But while the standard of excellence of the lower priced gas cars falls rapidly as compared to the $5,000 gas car, yet this is not the case with steam motive power. Its performance remains the same. So that from the commercial standpoint far greater advantage over competitors accrues to the steam car in the moderate price range than accrues in the high price range.

The owner of the moderate priced car runs it himself and keeps it in order, taking an interest in understanding it—an essential condition of success. The high-priced owner frequently does neither, and is consequently much more under the domination of his chauffeur, whose usual lack of knowledge of steam cars places the owner at a disadvantage. For this reason, again, the moderate priced car should have the better commercial opportunities. This is more particularly true because garage service is not yet developed for steam cars and dependence has to be made on the interested knowledge of the owner for overcoming troubles and accidents.

It is not my present purpose to dwell on details of construction except to say that these are receiving the closest study in several localities. The combustion system and boiler are receiving greatest attention, and it is safe to say that the near future will bring notable advances in these essential elements of the steam power plant. Determined interest is growing in the possibilities of the steam automobile, steam truck and steam tractor, not only on account of the serious fuel situation but because, whatever reason there was in the past for adopting the internal combustion engine when standards of performance were not as exacting as they are to-day, yet in view of these standards the steam plant now becomes the best way to meet them, and if standards further advance, may become the only way.

JOHN STURGES,
Platt Iron Works, Dayton, Ohio.

Novel Form of Universal Joint

Editor AUTOMOTIVE INDUSTRIES:

YOU will find enclosed a blueprint of a peculiar gear design (clipped from an axle assembly) which has come up in my work.

It is a gear of a spherical order used in a front or four-wheel drive on motor cars; when in the normal position there is no action between the teeth, they serving merely as a driving clutch; but when the front wheel is moved to the right or left, a rolling of the teeth in and out of mesh takes place. The working face of the tooth lies in a radial line from the center of the sphere.

It falls outside of anything in gear tooth curves that I have had any experience with. Can you give me any information as to how this tooth curve is generated or laid out and has there ever been a machine constructed in this country to cut a gear of this character; also, have you any idea along what line this could be accomplished in a manufacturing way? H. W. R.

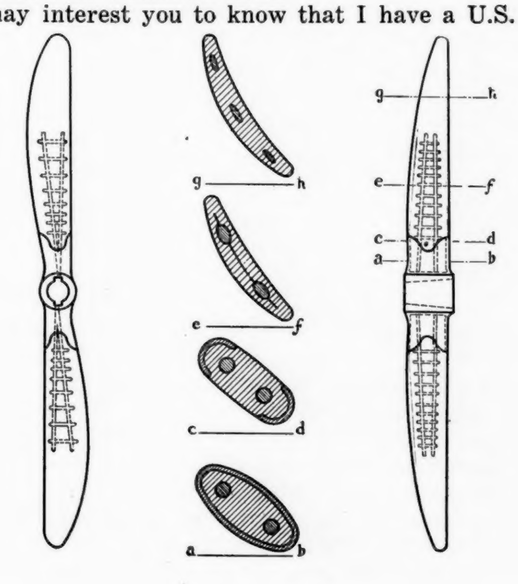
We have had no experience with this type of mechanism and cannot enlighten our correspondent regarding the necessary tooth form, etc., but possibly some reader can.—EDITOR.

Molded Airplane Propeller

Editor AUTOMOTIVE INDUSTRIES:

IN a recent number of AUTOMOTIVE INDUSTRIES, I see an article on a German aeroplane propeller built up of wood with steel reinforcements.

It may interest you to know that I have a U.S. patent



on an aeroplane propeller having a metal hub, with blades of moulded material reinforced with a metal frame work connecting with the hub.

My patent has nothing to do with the shape of the blades, but is basic as far as construction goes. That is, I can use any kind of molded material from soft rubber to hard rubber or bakelite. My type of propeller is, of course, not suitable for experimental work, owing to the expense of dies, but for quantity production of standard type, a most perfect and durable propeller can be turned out economically.

Hoboken, N. J. L. G. NILSON

Mr. Nilson encloses a copy of the patent referred to, which bears No. 1,308,527. It was applied for Sept. 28 1917, and issued July 1 1919. We reproduce one of the patent drawings herewith—Editor.

Combined Saw and Pump

Editor, AUTOMOTIVE INDUSTRIES:

IN your issue of Dec. 11, 1919, page 1157, we notice the erroneous statement that the Vauxhall company were the originators of the now well-known arrangement of mounting the fan and water pump on the same spindle and arranging the water pump immediately in front of the cylinder block and so getting a compact, simple and efficient arrangement. We now beg to point out to you that *we* were the inventors of this system, as is proved by our British patent, specification 20277/10, copy of which we enclose herewith, which forms the master patent of this device. You will readily understand that it is detrimental to our reputation when erroneous statements appear in an important publication like yours, and we shall esteem it a favor if you will kindly do your utmost to rectify the matter by giving due publicity to it in an early issue.

T. BLACKWOOD MURRAY,
Managing Director.

GLASGOW, SCOTLAND.

We have received the copy of the patent specification referred to, and wish to point out that this describes a combined fan and pump unit, the pump shown in the illustrations accompanying the specification consisting of an impeller which is located in the cylinder jacket space.—EDITOR.

Symbols Used in Gerster-Bradley Article

Editor AUTOMOTIVE INDUSTRIES:

WILL you kindly inform me of the meaning of the different literal terms of the formulas in the article, "The Design and Construction of the 183 cu. in. Engine," in the March 25 copy? For instance, the formula on the volume of charge drawn in per horsepower per minute:

$$V_1 = \frac{S^1 \times V_p \times 60}{n} = \frac{0.5026 \times 12.9 \times 60}{74} = 52.5 \text{ liters.}$$

I take it for granted that V_1 = the volume of charge drawn in, but I do not see what the other literal terms in the equation mean.

W. H. SINFIELD,
Portland, Oregon.

The symbols used in the article referred to are as follows:

V_p = piston speed in meters per second.
n = number of revs. p. m.
C = length of stroke in mm.
V_1 = volume of charge drawn in.
S^1 = area of cylinder head.
$f38$ = area of throttle passage of 38 mm. dia.
v = volume of compression chamber.
V = piston displacement of one cylinder.
p = volumetric compression ratio.
P = mean effective pressure.
N = brake horsepower.
t = number of strokes per cycle.
N_2 = number of cylinders.

—EDITOR.

Supercharging Aircraft Engines

Editor AUTOMOTIVE INDUSTRIES:

I WAS greatly interested in reading your editorial in the May 20 issue on "Methods of Supercharging." It has always seemed to me to be superfluous to have an extra or outside pump when the four-cycle engine is in itself, during two of its cycles, nothing but a compressor. Surely greater efficiency should be obtained by direct compression within the working cylinder than by outside compression with the necessary transfer of the charge.

In the early part of 1918, I suggested your remarks, almost verbatim, to the National Advisory Committee for Aeronautics, who advised me that "the supercharging booster or turbo compressor seems to offer the most promising results."

The method of control which I proposed in order to prevent too great a charge entering the cylinders at low altitude was by means of a duplex control consisting of an aneroid and a centrifugal governor. The latter would be driven from the engine in order that its speed be proportional to engine speed, and it would act upon an independent throttle above the carbureter similar to present governor control, but with the reverse action of the throttle.

Its influence upon this throttle would be based upon the volumetric efficiency of the engine over various speeds. The aneroid would also act upon this same throttle by means of a suitable linkage, and act either in conjunction with or contrary to the governor movement.

At low altitude the aneroid would tend to shut the throttle, allowing it to open more and more as the higher altitudes are reached. The centrifugal governor would close the throttle as the speed of the engine became less, so that in flying at low altitude the partially closed throttle would prevent the cylinders from drawing in a full charge. As the speed of the engine increases, the throttle would open to compensate for the loss in volumetric efficiency.

The throttle mentioned above is entirely independent of the manually controlled carbureter throttle. The aneroid and centrifugal governor would act on their throttle through a rod and linkage control, with the possibility of a cam movement interposed therein, as experiment might prove necessary.

AUSTIN M. WOLF,
I. Sekine Co., Inc.,
New York City.

Hub Multiplicity

Editor AUTOMOTIVE INDUSTRIES:

THE multiplicity of hub designs is a very vexatious problem for any engineering department of a wheelmaker, and at times we almost think that every engineer is trying to make the problems more difficult, instead of trying to simplify matters.

We are strong in the belief that there should be some united effort on the part of wheel manufacturers, and axle manufacturers, especially, to get together and devise some means of standardizing axles and hubs.

It took the European war and the United States Government to stir up the tire manufacturers in their standardization plans—for which we have to thank the kaiser—and if it had not been for this war, we really believe that we

would still have many more tire sizes than we had in pre-war times.

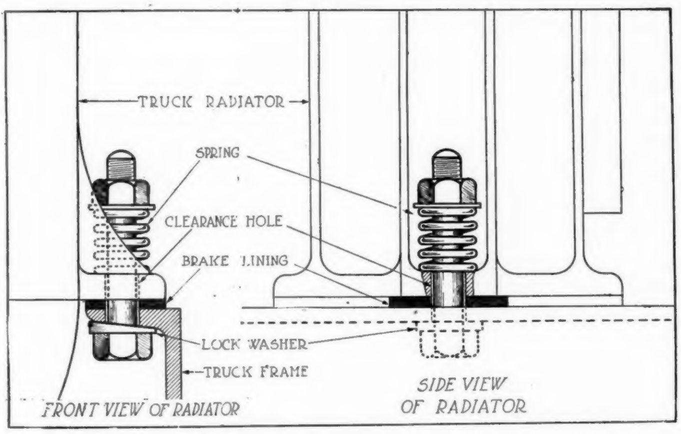
There is no doubt in the world AUTOMOTIVE INDUSTRIES can do a whole lot of good toward getting axle manufacturers together on this point, and you will have our heartiest co-operation. You can call on us at any time to help you in reaching the goal of a standardized axle.

J. S. CULP, Chief Engineer,
The Reliance Wheel Co., Youngstown, O.

Mounting of Radiators

Editor AUTOMOTIVE INDUSTRIES:

THE accompanying sketch shows a method of mounting the radiator, which the writer planned in 1912 and incorporated in an experimental model in 1913. As will be seen, the side members of the radiator have feet which rest on a piece of brake lining, or other relatively soft material which will absorb vibration, this material in turn resting upon the top of the frame side rail. The radiator is steadied in this vertical position by a tie bar of the conventional sort from the top tank to the dash. The holding-down bolts through the feet on the side pieces of the radiator are long enough to take coiled springs shown.



A method of truck radiator mounting by which the radiator may be relieved of distortion due to excessive frame weave. The spring is not used to support the radiator but to lessen the vibration

The bolts pass through holes which have considerable clearance, both in the radiator foot and the frame rail, and, in connection with the spring and the soft pad, allow for a large amount of frame distortion without straining the radiator in any way. I note that a great many builders are now adopting this construction, which I think has great merit. I incorporated it in the design of the Fageol trucks, and on the original layouts of the Class A and Class B military chassis, which were made in my office in Detroit. This type of mounting is much superior to one in which the radiator is supported on a spring, for with that construction there are times when a very pronounced vibration is set up in the entire radiator, due to the period of the supporting springs synchronizing with the vibrations of the engine or the road springs. With my design all such vibration is damped out by the supporting pad, and the function of the retaining spring is that of allowing the radiator foot to rock on the soft pad, without stressing and distorting the radiator parts.

I notice that the following makes of trucks are using this construction: Koehler, Paige, Norwalk, Gramm-Bernstein, Autocar (new 3½-ton), Brockway, Packard, Traffic, Maccar, Huffman, Sterling, Service, and Indiana.

CORNELIUS T. MYERS.

A Suggested Means for the Elimination of the Exhaust Cutout

Editor AUTOMOTIVE INDUSTRIES:

SOME years ago I was experimenting rather crudely with an exhaust installation for a small launch with a one-cylinder, two-cycle engine, which was exactly suited for the boat. Among my experiments was one by which the exhaust was led directly out under the water. I found the back pressure prevented starting, although when the boat was in motion it would exhaust under the water with better effect than into the air, actually turning over more revolutions, although with considerable jarring of the boat.

My experiments to determine how much of a cutout was necessary for starting showed that as against a 1¼-in. exhaust pipe, a ⅛-in. opening was amply sufficient. This gave me the impression that a materially smaller opening would have worked just as well.

I do not recall that there was any noise worth speaking of from this source. It has occurred to me that it might be worth while for those who have the time to experiment, to find out whether several minute openings, possibly 1/16 in. in diameter, drilled in the exhaust piping, 8 or 10 in. in front of the muffler, would not provide against all the back pressure which is found in an automobile exhaust.

There will also be the question as to whether flame might pass through these openings in a way to be dangerous. Barring that, and if the openings do as I think they probably will, neutralize the back pressure, it would seem unnecessary to provide cutouts, at least on the car of average price, as well as upon all trucks. In fact, as to trucks, a continuous slight puffing noise from such openings would be quite unobjectionable. Beside slightly diminishing the cost of the car, the main advantage would be to take from the driver his present opportunity of making a nuisance of himself.

Many drivers do not seem to understand that the cutout is very rarely of any use outside of racing cars. Back pressure only develops at full throttle opening.

Aside from those who seem to enjoy the racket, there are many who seem to make a conscientious practice of opening the cutout whenever they start the car or whenever a slight grade is before them. Viewed as an attempt at intelligent driving, it is amusing to see the driver of an empty truck using his cutout on a 3 per cent grade upon a smooth city street. I have even seen the driver of a Stutz runabout open the cutout on such a grade with no one but himself in the car.

If the experiment proves successful, the law might be invoked to prevent cutouts capable of being used from the driver's seat. They have a limited use in testing out the working of the cylinders, but a little inconvenience on those comparatively rare occasions might be put up with in exchange for the general diminution of noise which would result.

CHARLES E. MANIERRE.
New York City.

Books and Circulars Covering Automotive Subjects

THE Iron Age Catalog of American Exports is designed to give the manufacturers, engineers, importers and other buyers in all countries a compendium of readily understandable data of the products manufactured in this country. In order that the foreign buyer may be adequately advised of products which American manufacturers have to offer, it is written in five control languages of the world, English, Spanish, French, Portuguese and Russian.

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Our Engineering Number

IN the present issue, our Engineering Number for 1920, engineering problems of the automotive industry are being covered in a rather comprehensive way. There are articles pertaining to passenger vehicles, motor trucks, tractors, aircraft and motorcycles; then there are articles on metallurgical subjects, ignition, lubrication, fuels, carbureters, fuel distribution, electric generating apparatus, engine research, etc. In preparing the issue this year, some difficulty was encountered at first, owing to the fact that engineers generally are passing through an exceedingly busy period; nevertheless, we succeeded in getting together a plentiful supply of interesting material, so that we even had to hold over some articles for future issues.

Every effort has been made to avoid errors, but it is hardly to be expected that in an issue of this size, dealing with many involved subjects and containing a good deal of mathematics, they should have been entirely eliminated. We believe that the issue will be appreciated, especially by readers of a technical turn of mind, who will probably keep it for reference for a long time.

Solidarity in the Engineering Profession

THERE is now a movement on foot to bring about closer relations among the engineering societies of the country, and a convention to further this was held in Washington June 3 and 4. What is really aimed at is a federation of all great engineering organizations, to the end that there may be solidarity in the profession and that it may be of the greatest service to the city, state and nation.

The different engineering organizations co-operated almost for the first time during the war, when they joined their efforts in helping the Government. One of the outcomes of their collaboration then was the organization of the Engineering Council and the American Engineering Standards Committee.

There has been a growing class consciousness among engineers in late years, and the war, in which the work of the engineer counted for so much, served to accentuate this. The different branches of engineering—mechanical, electrical, civil, mining, automotive—already are well organized, but there is as yet no organization to represent the whole engineering profession.

It is undoubtedly one of the objects of the movement to raise the standing of the engineering profession generally and to teach the public to better appreciate the services of the profession to the community. Engineering enters into the life of organized society in so many ways that it is necessary to have men with a thorough engineering knowledge attached to all the various governmental bodies, and a general knowledge of the principles underlying engineering work would be a great help to those in the highest governmental positions.

There has been a similar movement in Germany, which started some years before the war and is being carried on with increased vigor at the present time. One of the complaints of German engineers has been that they are not eligible to the higher administrative positions, such as that of city mayor, one of the conditions attached to the candidacy for which is a legal education. German engineers hold that success in the administration of a large city demands a broad knowledge of science and industry even more than legal knowledge, and that if the course of education of the candidates is to be prescribed, an engineering education should at least be put on a par with a legal education.

Administrative or executive ability, of course, is a thing that is more or less inborn and not dependent solely upon the course of training received. It is probably found as much among engineers as among the members of the legal profession. In the past, engineering has been defined as the art of controlling the forces and utilizing the materials of nature for the benefit of the human race, but in view of the tremendous increase in the scale on which engineering work

is carried out it has been found advisable to amend this definition, and in addition to the above it is now said to include "the art of organizing and directing men."

The engineer certainly wants to utilize his knowledge and skill for the benefit of the public, and he wants the fullest opportunity to do so. The engineering profession is broadening its view of life and is placing among its ideals unselfish devotion to the city, the state and the nation.

Valve Cam Forms

IN gas-engine design three different forms of cams have been commonly used so far, the tangential, the convex flanked type used in conjunction with a mushroom cam follower, and the constant acceleration type. Each of these is subject to variations even if the valve timing and the lift are fixed, so that a great variety of lift curves are possible. The constant acceleration type has generally been regarded as a very suitable cam for high speed engines, for the reason that with constant acceleration the strain on the valve mechanism during the acceleration period is constant and there is no excessive momentary strain which might cause breakage. Therefore, with a constant acceleration cam the valves can be opened in the least possible time with a minimum danger of breakage. It is this consideration which is determining in racing and aircraft-engine design, and constant acceleration cams have been used mainly in these engine types.

In passenger car engines, while we are also aiming at high speeds of revolution, we have to insure silent operation, and in this respect the constant acceleration cam is not so very satisfactory. This will be evident when it is considered that with this type of cam the maximum lifting force is applied instantly and removed instantly. The cam follower, therefore, is virtually being struck a hammer blow and it is quite obvious that this sudden application of the lifting or accelerating force must result in noise. It would certainly be much better, theoretically at least, to increase the acceleration gradually from nothing to the maximum, instead of applying the maximum instantly. This does not necessitate a slower lift, as the maximum rate of acceleration could be raised.

A good plan for laying out the cam would seem to be the following: Let the acceleration increase at a uniform rate from zero to the maximum; let it remain at the maximum for a certain period, then let it change at a uniform rate from the maximum positive to the maximum negative value, which latter should be less than the former, so as to obviate the need of exceedingly stiff springs; after remaining at the maximum negative value for a period, let it decrease to zero at a uniform rate.

The difference between a constant acceleration and a gradually increasing acceleration is well illustrated by a comparison of the way in which curves are negotiated by automobiles and railroad cars respectively. In the automobile the driver swings his steering wheel around gradually, so that the radius of curvature of the path described by the car decreases gradually from infinity as the car leaves the straight course. As a result there is no sudden lateral acceleration and side sway; on the other hand, in laying out tracks on railroads, a curve of finite radius joins the straight portion directly, and the lateral acceleration therefore instantly assumes a definite value. The unpleasant sensation, if the speed at which the train goes into the curve is at all high, is undoubtedly familiar to the reader.

Unfortunately the refinements which can be made in valve mechanism as above described are largely vitiated by the variation in valve clearance with changes in the working temperature of the engine. To obtain a really quiet engine it is necessary to reduce the dependence of the clearance upon engine temperature to a minimum.

Push Standardization Work

WE must plan the automotive industry for what it will be ten years from now. Big as it is now, it is only an infant to what it will be then. The motor truck is just beginning to come into its own. The railroad difficulties have shown what the truck can do with the poor roads that we have to-day. When the roads are better and trucks are more improved, they will go much further in supplanting the railroad for short hauling.

To make the industry all that it can be, the need of standardization is great. It is not right that every truck body builder should be handicapped by lack of standardization in the chassis upon which the body is mounted. It is not right that the wheel maker must suffer for the originality of axle engineers when nothing is gained by minute variations in hub dimensions. Power take-offs should be standard on all truck gearsets, in order that dump body manufacturers and others will be able to provide their customers with suitable equipment at the least possible cost and to operate at the greatest possible efficiency.

The industry as a whole should insist on standardization where lack of it puts a penalty upon other branches of the industry and results in raising the cost of the product to the ultimate consumer. Standardization is the greatest implement in the hands of the manufacturer to make it possible to reduce the cost of car, truck and tractor. It should be used judicially but persistently so that it will not be necessary for one branch of the industry to tie up millions of dollars in tool equipment to co-ordinate itself with the vagaries of another branch.

We have done more in this country than in any other toward standardizing out automotive equipment, but we have only scratched the surface. There is hardly a branch that could not mention a score of points which if standardized would greatly reduce the cost of manufacture and eventually reduce greatly the money tied up on the part of dealers in dead and semi-dead stock upon the shelf. Eventually it is the consumer who pays for this, but the industry suffers because the volume of output is directly affected by the selling and maintenance cost.

Detroit Nearing Normal Stride

Production in May Tribute to Industry

Month Shows Gain of 60,000 Despite Unusual Handicaps—Trucks Swell Output

DETROIT, June 5—Despite unfavorable financial conditions, continuation of the railroad strike and other handicaps, automobile and truck builders in the Detroit district increased production in May considerably over the figures for April, in some cases more than doubling the April output. Figures from all factories in Michigan and including the Willys-Overland plants in Ohio, compiled from statements of executives, show a total of 156,178 cars and 19,532 trucks manufactured during May. This compares with 111,961 cars and 15,771 trucks in April.

A note of optimism is apparent throughout the district. Factory executives who in some instances took a very pessimistic view of the situation three weeks ago and some of whom even went so far as to express fear of a shutdown, to-day are enthusiastic over the month's returns and predict a still better record for June.

There was a slowing up in production a few weeks ago, due chiefly to freight congestion and the inability of the manufacturers to get parts and materials sufficient for normal output. The natural result was reflected in the laying off of men in each factory, though the total was nothing like the number erroneous reports have made it, and at no factory did the order affect more than 15 per cent of the employees.

Remove Incompetent Help

Instead of hampering production that action appears to have had an energizing effect on the plants and in the opinion of a majority of executives served only as a means for removing incompetent and inefficient employees. Naturally the many reports and rumors of panic coupled with the laying off of some employees in each factory had the effect of stimulating to renewed energy the other employees, and it can be stated truthfully that labor in Detroit to-day is producing more nearly on a 100 per cent basis than has been the case in the last three years.

There is no indication that any factory has been so badly handicapped by steel shortage as to cause fear of serious interruption to production as a result. The freight situation is easing up considerably and the efforts of the manufacturers appears to have been crowned with success in the light of the prompt action of the government authorities. Materials are arriving in better quantities and practically all factories now are getting at least a few cars each day for the delivery of their finished product.

Tribute to the truck as an aid to the crippled transportation systems during May cannot be too strong. But for the help of the trucks many factories would have had to close. The truck caravans still are going back and forth to junction points and many of them to the supply centers to bring in material. This adjunct to the railroad system will be maintained and increased in the effort to speed up the factory output.

Ford Makes Big Gain

Ford Motor Co., employing an army of men, can be taken as the best example of conditions for comparison. Ford in April built 51,066 cars and 7,179 trucks. In May the Ford factory turned out 70,000 cars and 10,000 trucks. This in the face of reports circulated widely throughout the country that the Ford factory had laid off from 10,000 to 20,000 men. As a matter of fact, between 600 and 700 employees, confined entirely to one department of the company's plant, were given an "indefinite vacation." These employees were given the alternative of a transfer to the other departments in the factory and a majority of them availed themselves of that privilege. The others accepted what wages were due them, looking upon the "indefinite vacation" as a discharge. This move on the part of Ford was simply to increase plant efficiency and that it proved highly successful is demonstrated by the production record, despite the handicap of freight car shortage. Ford has set 100,000 as the output for June, and with the railroad situation improving steadily it is likely the figures will be reached.

Similar conditions existed in all of the factories, notably Hudson-Essex, Cadillac, Chevrolet, Dodge, Hupp, Paige, Reo and Overland.

Weather Aids Driveaways

The good weather was an invaluable aid to the manufacturers in the opportunity it furnished for driveaways. The warehouses, barns and other storage places that have been filled with cars rapidly are being emptied and the cars sent on to the dealers by the railroads where necessary, or by driveaways. The driveaway has furnished an ideal opportunity for dealers to bring their customers to the factories and make the visit a family party, with the owner's family and friends making the return trip home in their own car. These family drive-aways have been encouraged by the factories, all of which through their dealers have extended an invitation to owners to bring their families

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Mexico in Market for Automobiles

Stabilizing of Conditions Expected to Lead to Wide Demand for Cars

MONTEREY, MEXICO, June 7—With the restoration of peace in Mexico, which condition is generally regarded as more assured now than at any time during the last ten years, it is expected that a wonderful increase in the automobile trade in this country will take place. Along with the probable development of this business will come a great increase in the sale and consumption of gasoline and lubricating oils.

In anticipation of the early opening of trade along these lines several of the larger American companies that are engaged in the marketing of petroleum products are preparing to build filling stations in all of the larger cities of Mexico. At this time there is a great need for such adjuncts to the automobile traffic, it is asserted. If the plans of the companies are fulfilled scores of modern filling stations will be constructed in Monterey, Tampico, Saltillo, San Luis Potosi, Chihuahua, Torreon, Guadalajara, City of Mexico, Vera Cruz and other cities.

For several years Mexico has been the dumping ground for thousands of second hand or used automobiles from the United States. Compartively few new automobiles have been shipped into the country, except to Tampico, where conditions have been more or less normal during all of the revolutionary period. The country is well-stocked with these old models and somewhat dilapidated cars. With the assurance that tranquility has come to Mexico, and with the restoration generally of commercial and industrial enterprise the demand for automobiles, motor trucks and delivery vehicles will show an enormous increase, it is believed. Already a number of new agencies have been established in the larger cities and they are awaiting the advent of more stable conditions to open up business on a large scale.

To Improve Road System

Good road construction also promises to receive much attention on the part of the several State governments and the Federal Government as soon as political affairs are well straightened out. In the Tampico region already a system of modern highways has been built by American and other foreign oil companies for the especial purpose of solving their own transportation problems, both as to the use of automobiles and motor trucks. A general road policy following the Tampico system is to be evolved.

Fear of Deflation Period Passed

Business Editors See Good Times Near

Sound Industrial Policies Overcome Possibility of Depression—Must Watch Credits

CHICAGO, June 5—Business conditions throughout the country were discussed at the National Conference of Business Paper Editors held here yesterday. Speakers representing various lines of industry declared constructive optimism was essential to the well-being of the nation and it was their opinion there was no occasion for depression on the part of those prepared to meet the changing conditions resulting from readjustment and deflation of credit.

In spite of the many handicaps to which it has been subjected, there is every reason to believe steel production for the year will be on an 80 per cent basis. It is certain there will be no unemployment unless it is voluntary on the part of workers. There are rumblings of another strike, but the only solid basis found for these reports is the fact that steel workers in the Pittsburgh district are saving their money, which is so unusual that it is causing comment.

The railroads are gradually untangling the traffic jam, and conditions at the terminals have improved rapidly in the last few days. This has led to material increases in the stocks of supplies being laid down for manufacturers. One striking point brought out is that the railroads carried more freight in April than they did in the same period last year in spite of labor troubles and congestion on the rails. It will be three or four years, however, before the carriers can handle all the business which will be offered them, and shippers will be beset constantly with all kinds of difficulties.

Business men in all fields of activity are finding credit the most difficult problem which confronts them. Those who are on a solid foundation are being given the banking assistance absolutely essential, however. Many of them are inclined to accept too many trade acceptances, and they find it difficult to cash in on them at the banks. It was stated there seemed to be a widespread misunderstanding of this variety of paper, and all business men were advised to study the subject carefully.

Careless Methods Must Go

Carefully prepared and accurate financial statements never were so essential as now to men seeking bank credits. The old haphazard methods will not do with credit strained as it is now. Employment of expert auditors was advised.

Building supplies are dormant and prices are falling, but demand is expected to reassert itself when new and stable price levels are reached. In this field many mills will have to suspend unless their sales increase. The decline of prices is not expected to stop until they approach the cost of production. There is no reason to fear exhaustion of timber supplies, for at the present rate of consumption that now in sight will last half a century.

The message from the farms was to the effect that stories of alarming reduction of acreage with impending dangerous crop shortage were greatly exaggerated. It was believed there would be bumper crops of many food products. There is a shortage of about 17 per cent of farm labor as compared with last year, and 30 per cent compared with the pre-war period.

General Markets Hard Hit

The markets for shoes and leather products are virtually flat because of the absence of orders which normally are in hand at this period. This situation results from the backward season, railroad delays and the refusal of consumers longer to pay high prices. Price reductions in this field and in dry goods were described chiefly as a move to bring about stock turnovers. It was said dry goods merchants could find nothing in the wholesale market to justify the lower scale of prices now in vogue in many cities.

In connection with foreign trade it was said the exchange rate with England could be corrected by the government at almost any time if it desired to do so. It has resulted in building up an artificial trade barrier, and British subjects all over the world are having drummed into their heads the motto, "English goods for English subjects."

Nothing Serious Anticipated

In general, it was believed there is no prospect of serious business depression or curtailment of production if manufacturers proceed along sane and cautious lines. It was conceded there would be a slump of uncertain duration and dimensions in almost every line, but it was felt there would be goods and credit for every one honestly entitled to them. Poor business men and those who have not built on a solid foundation are facing a dangerous period, however. Speculation and expansion under present conditions was frowned upon.

The conditions which now prevail in the automotive industry are found generally in almost every other field and there was no indication of a disposition to discriminate against any particular line. The chief reasons for pessimism at present are uncertainty about credits and prices. There was an enthusiastic and unanimous belief that once a condition something akin to normal is restored the nation will be blessed with a long period of prosperity.

British Lose Hold on Colonial Trade

South Africa Leaning Toward American Cars, Makers Warned —Propose Subsidy

LONDON, May 21 (*Special Correspondence*)—A. R. Atkey, M. P., ex-president of the Motor Trade Association, and member of a prominent firm of British and Colonial motor dealers, contributes to the London *Field* some notable remarks concerning the South African market for cars, and in particular the position and possibilities of British trade in that territory. He points out the great importance of the South African market and says that British trade has ill-requited the hopes and preferences of the South African for British cars who put up with disadvantages, such as insufficient axle clearance.

He still has a patriotic preference for British cars, and on the armistice being declared began to get busy with orders sent to Great Britain but with "most disappointing results." Since then he has been unable to fill his orders for British cars and has seen their cost soaring higher and higher, and much greater than the advanced cost of American cars. To-day Atkey says the South African outlook is "distinctly unfavorable to British manufacturing interests."

He suggests that the British car makers should in effect *tax their home car buyers from* $500 *to* $1,000 *per car, out of which to subsidize the colonial motor trade.* He finishes with the remark that "there is a limit to which the colonial buyer will go in the way of payment for his loyalty to British interests, and unless the home manufacturer can compete reasonably in price with the American, or the foreign made article, he will fail to realize the great asset which is available in the South African market."

Atkey justifies his subsidy proposal on the score that the British car buyer is freely paying premiums on cars and indirectly encouraging private profiteering. Apparently Atkey is counting on the continuation of the 33 1/3 per cent tariff on American cars, operating to facilitate this subsidy scheme, at the already hard-hit Britisher's expense, but this fact illustrates very aptly the truth so much insisted on by free-traders, that the tariff chiefly operates to injure rather than protect the people it is intended to benefit and protect.

STRIKE AT ROLLS-ROYCE

LONDON, May 21—(*Special Correspondence*)—A strike of some 5000 men at the Rolls-Royce Works, Derby, is under way to bring about the reinstatement of a shop steward.

Square Deal Sought on Publicity Rule

Detroit Advertising Managers to Urge Modification of Publishers' Policy

DETROIT, June 8—Advertising managers from all the automobile and truck factories in and around Detroit at a meeting to-day named a committee of six, headed by H. C. Dart of the Paige-Detroit Motor Car Co. to co-operate with a similar committee from the New York Automobile Dealers' Association and other committees, which will seek modification of the ruling of New York newspapers eliminating automobile publicity. Short talks were made by several of the advertising managers, the burden of all of which was that revenue derived from automobile advertising is the third largest in the industrial world, in view of which fact it was contended that the automobile industry was being discriminated against in favor of picture shows, theaters and other industries.

H. T. Gardner, executive secretary of the New York association, came to Detroit to explain to the factory advertising managers what he termed was the unjust attitude of New York dailies, publishers of which have issued a hard and fast ruling prohibiting the mention of the name of an automobile or an automobile company in the news columns. He urged that the factory executives co-operate with the National Automobile Chamber of Commerce, the accessories and parts manufacturers and allied interests in the effort to have the ruling rescinded or modified. Ward Canaday of the Willys-Overland Co. said the situation was similar to that which existed in Boston a year ago, which had been changed as a result of the combined efforts of automotive manufacturing executives and dealers, and contended that a dignified and truthful presentation of all the facts would suffice in the present instance to secure at least a modification.

All of the manufacturers admitted they were more or less to blame in the amount of purely advertising matter which had been dumped into the newspaper offices as publicity, and they were a unit in admitting the justice of the attitude of editors in throwing out much of this propaganda, their only grievance lying in the fact that they felt the publishers had gone to the extreme in eliminating all mention of the name of factories or cars.

Advises Against Quarrel

C. A. Brownell, of the Ford Motor Co., urged a "go-slow" policy, and contended it was the best plan not to quarrel about other industries, but to show the merits of the automobile industry and logical reasons why so many millions of people are interested in news of the industry and the different makes of cars. Brownell drew a sharp line between news and advertising, and declared the reason prompting the publishers to a great extent was the purely advertising matter sent out as news. He said the word publicity was overworked, and that advertising managers had come to regard publicity as meaning simply to tell a story about the wonderful performance of their particular car, which was of interest chiefly to the manufacturer and dealers. The gist of his statement was that the publicity matter sent out is written too much like patent medicine advertising and has no call upon editors for publication in news columns.

Committee to Present Views

In making the motion for the appointment of the committee, Gordon Muir of the Maxwell-Chalmers organization said it was the consensus that the automobile industry should be given an even break in the matter of free publicity in connection with paid advertising. Dart, who was chairman of the meeting, named Canaday, Muir, Cliff Noble of the Liberty Motor Car Co., L. B. Dudley of the Federal Truck Co., and W. A. Holmes of the Packard Motor Car Co., with himself, as the committee to draw up a resolution which will be submitted at another meeting of the advertising managers outlining the views of the factory executives. Meanwhile, the independent advertising agencies will be asked to name a committee to co-operate.

Truck Tour Postponed; Rail Congestion Cause

OMAHA, June 5—The first national motor truck reliability contest which was to have circled the "money belt" of the Middle West has been indefinitely postponed. The impelling reason is that manufacturers have found it would be impossible to get trucks to Omaha in time for the starting date without driving them there and it is felt this would impose an unfair burden on those in the East. The promoters of the run feel it would be better to wait until there is a marked easing in the material and transportation situations.

ANALYZES RAILROADS AND CARS

AKRON, June 9—The B. F. Goodrich Rubber Co. has made an interesting analysis of railroad and motor vehicle figures which shows a total highway mileage of 2,478,552 and a total railroad mileage of 253,626 in the United States. Operating over these mileages are 7,596,424 motor vehicles and 2,439,206 railroad vehicles. Taking the entire country, there is an average of 59 yd. of railroad to each motor vehicle in operation. The figures include both cars and trucks.

COTTAGE FOR GIRL CLERKS

JEANNETTE, PA., June 5—In an endeavor to help solve the vacation problems of the young women employed in its offices, the Pennsylvania Rubber Co. has leased for the summer season a commodious cottage at Put-In-Bay, Ohio, and turned it over to the girls, twelve of whom can be accommodated at one time.

Motor Truck Lines Spring Up in Texas

Merchants and Manufacturers Impressed With Transportation Possibilities in State

AUSTIN, TEX., June 4—During the last few weeks motor truck runs or excursions have been made from several of the larger cities of the State. Some of these trial trips covered distances of several hundred miles and were made under extraordinarily trying conditions as to roads. The special purpose of these motor truck runs was to demonstrate the practicability of using this method of transportation under all kinds and conditions of weather and roads.

The ship-by-truck movement has shown a remarkable development in Texas during the last several months. This is said to be due largely to the shortage of railroad cars, or at least to the unsatisfactory movement of freight by rail. Running out of Dallas, Fort Worth, Austin, San Antonio, Houston and other towns are regular lines of motor trucks which carry shipments to many towns adjacent to those respective commercial centers.

Among the recent test motor truck trips were those which were run out of Dallas, Fort Worth, Wichita Falls and Amarillo. The motor trucks were given the severest possible usage, in some cases negotiating long stretches of muddy road, crossing many running streams, mounting steep grades and otherwise being put to a thorough test as to stability and endurance. It is stated that the merchants and other business men of the rural communities which were visited on these motor truck runs were especially well pleased with the success of the tests, and that the trials promise to lead to the early establishment of many regular motor truck shipping routes.

TIMER COMPANY EXPANDS

MILWAUKEE, June 7—The Milwaukee Auto Engine & Supply Co., considered the largest maker in the country of Ford and Fordson timers, and also manufacturing safety guards, bumpers, etc., has increased its capitalization from $100,000 to $200,000. A new plant with 15,000 sq. ft. has been completed at North Avenue and Thirtieth, in which last year's output of 500,000 timers will be increased to 1,250,000 in 1920. The old works at 837-841 Twenty-ninth Street will be devoted entirely to bumper making. It has 12,000 sq. ft. B. D. Zimmermann is president and general manager.

ACME TRUCK PRICES RISE

CADILLAC, MICH., June 5—Acme Motor Truck Co. has increased the price of 1, 1½ and 2-ton models $100 each, the prices now being respectively $2,175, $2,375 and $3,030. The 3½ and 5-ton models have been increased $175, making the prices now $4,050 and $5,150.

Car Credits Grow Sixfold in Year

Texas Bankers Tell Association Deflation of Money Market Is Imperative

CORPUS CHRISTI, TEX., June 5—At a recent meeting of the Corpus Christi Automobile Trades Association the subject of extending credits to the automobile industry was discussed. The fact that certain industries had recommended to the Federal Reserve Board that the automobile industry was non-essential and therefore not entitled to credits made the subject of timely interest. The discussion was led by W. R. Norton, vice-president of the City National Bank, and Joseph Hirsch, president of the Corpus Christi National Bank.

"Credit extensions for the month of April, 1920, were 600 per cent greater than the corresponding month last year," said Norton. He made the statement that a great many people bought automobiles who were unable to buy them, and this condition augmented the present condition of inflated credit. "Banks were borrowing heavily from the Federal Reserve Bank to carry credits," he continued. "Realizing the outcome, unless the granting of credits was curtailed to some extent, the Federal Reserve Banks advised various banks to bring about a deflation, by making only loans that were productive. A letter received later said that the suggestion previously sent would be put into force by instituting a sliding scale of rates. The sliding scale was invoked to decrease credits by placing a higher rate on loans above a certain amount.

"The recent financial panic in Japan was due to inflated credit. It is better now to call a halt, says the Federal Reserve Board, so that when prices go down the people can take care of themselves. The demand for money is due to the expansion of business and the general reconstruction period brought about by the war. The situation now confronting us will be worked out and without a money panic. The Federal Reserve Board is the most powerful financial machine in history, and will work out a satisfactory solution of the money situation."

Larger First Payment Required

Joseph Hirsch said that he saw no reason why the automobile dealers should be alarmed. An open discussion brought out the fact that automobile finance corporations still were taking paper, but were insisting that a greater amount be paid down when the car was purchased. Some dealers also have been advised that a higher brokerage would be put on automobile paper.

"Don't cross a bridge until you come to it," said Hirsch. "Of course, if the finance corporations refused to take paper it would be difficult for the banks to take it after it had been refused. The money market is not as stable as it was, but I see no cause for alarm. We have had very little trouble with automobile paper we have handled, I am glad to say."

In commenting on the fact that the automobile had been classed by some as non-essential, and therefore not a subject for credits, Hirsch said he believed the attack had been unfair.

Milwaukee Parts Plants Restrict Production

MILWAUKEE, June 7 — Milwaukee, as one of the principal centers of the automotive parts industry in the United States, is feeling the effects of the reduction in the production program of passenger car manufacturers of the country. Since June 1, operations in numerous large plants are gradually being restricted, production declining to 60 or 65 per cent of capacity, which up to this time has been held variously at from 85 to 100 per cent, dependent upon supplies of raw materials and ability to make deliveries through the railroad traffic tangle. The release of men is being accomplished gradually without upsetting conditions. The situation of the automotive parts industry is not different from several other principal industries in Milwaukee.

STEEL COMPANIES TO MERGE

WHEELING, W. VA., June 5—Plans for the merger of three of the largest independent iron and steel corporations in this section of the country are about completed and the incorporation of the three probably will be announced before July 1. The corporations included in the merger are: Wheeling Steel & Iron Co., La Belle Iron Works and Whitaker-Glessner Co. They will make a corporation of approximately $100,000,000 capital stock.

Albany to Establish Motor Line Depot

Terminal Point for Freight and Passenger Service Planned by Chamber

ALBANY, N. Y., June 4—The Albany Chamber of Commerce is taking steps to establish a terminal station at a central point for automobile bus and motor truck freight and passenger lines. There are from 35 to 40 motor freight lines operating out of Albany and almost as many passenger lines, with no central terminal from which they are operated.

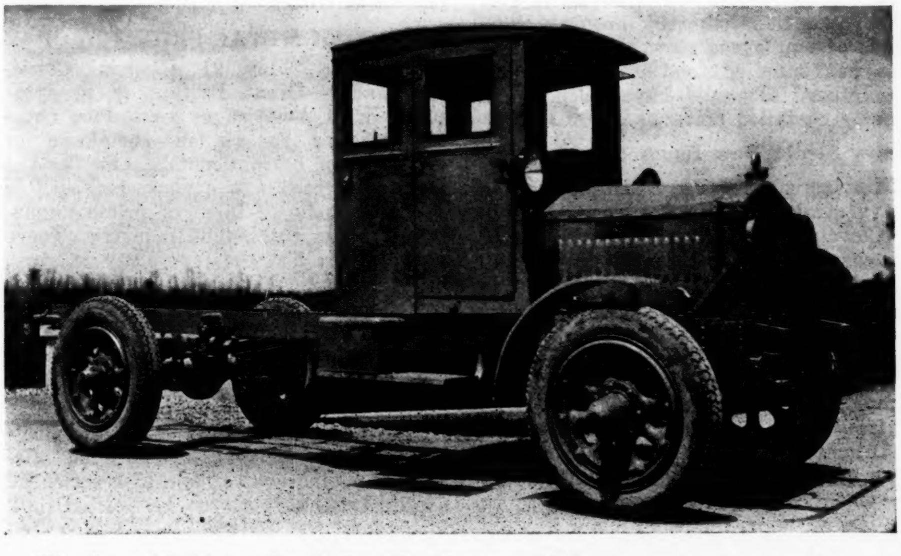
Cars on all these lines leave from different places at different times, with the result that the shippers are confused and often uncertain where they can secure the particular truck or bus they are looking for. With the new plan, goods sold by Albany wholesalers, retailers and manufacturers to be delivered along the lines of the bus routes may be taken to the terminal and there handled by a company organized to arrange shipments. Many passengers would use the bus that do not at the present time if they could go to the central terminal, and truck shipments could be arranged for in a more satisfactory way.

A site for the station is to be selected very shortly and all arrangements made.

COMMERCE MODELS ADVANCE

DETROIT, June 5—Commerce Motor Car Co. to-day announced an increase in the list price of model E chassis $165 and on model E. P. $190. The increase will make the list prices as follows: Model E. solid tires, $1.855; model E. 35 x 5 pneumatic cord, $1990; model E. P. 36 x 6 pneumatic cord, $2,255.

One of the Reynolds Pneumatic Line



The Reynolds Motor Truck Co., Mt. Clemens, Mich., is getting into production with a line of seven trucks. Four are for solid tires in 1½, 2½, 3½ and 5-ton sizes, and three are pneumatic models, 1½-2, 2½-3 and 3½-ton sizes. The pneumatic tire models have a power-driven pump as standard equipment

Overloaded Trucks Menace Ohio Roads

Officials Act to Prevent Ravages—Truck Men Blame Highway Construction

CLEVELAND, June 7—Truck manufacturers and dealers, truck owners and officers of automobile clubs will meet in Cleveland in the near future to plan a campaign to preserve improved roads. Announcement that the meeting would be held was made after the Cleveland Automobile Club unloosened its heavy artillery and fired a round of shot at overloading and speeding on the part of a few truck owners engaged in motor transportation of freight.

Fred H. Caley, secretary of the club, made it plain when he fired his first broadside in a letter to Cuyahoga county commissioners, that he was aiming to help motor transportation of freight by putting out of business a very few truck drivers who are ruining improved roads in northern Ohio.

"It is wrong that the great majority of men engaged in transporting freight by motor truck should suffer from the acts of a very few, who not only overload their trucks but then speed them," said Caley. "The great majority of drivers do not load up too heavily and they want to see improved roads kept up. The very few, on the other hand, who overload seem to have no thought for the future. They are getting all they can now, getting it as fast as they can, overlooking the fact that they are ruining improved roads and thereby injuring chances of the legitimate truck owner, who obeys traffic laws."

The tie-up of freight on account of the switchmen's strike brought about an extraordinary increase in interurban trucking. Motor trucks in the past two months saved the situation for cities and villages in Ohio. The trucks by bringing in raw supplies enabled factories to continue operations, thereby keeping thousands of employees at work. The truck also has supplied food for thousands, who otherwise would have been on a greatly curtailed diet.

Heavy Strain Now Imposed

The increased traffic naturally laid a heavy strain on improved roads leading to Cleveland and Akron especially. In one case brought to the attention of club officials a truck driver was reported to have driven a load of approximately 25 tons over a brick pavement, the weight of the load sending the rim to the bricks and crushing some of them. Another complaint is the driving of 17 to 19 tons of freight over the improved roads.

Automobile dealers and truck owners were quick to offer their support to the movement so long as it was directed to the law violator, but they made it plain that the burden for the breakdown of roads cannot rightly be placed on truck drivers. These men attributed the breakdown of the roads to two factors: First, that road builders years ago failed to visualize the future of highway transportation and made roads too light; second, failure on the part of public officials to limit any or all traffic over the roads during spring thaws.

The experience of the last three months, they said, had shown these facts: That interurban transportation by truck has come to remain and that from this time on remedial measures alone will not be sufficient; that new policies in road building must be adopted at once to meet the necessities of truck traffic.

Leading road engineers and road traffic experts of the country will be asked to attend the conference. Not only will the technical problems of road building be discussed at this meeting, but the problems of financing and maintenance.

Horseshoes Lucky for Trucks

ST. LOUIS, June 7—Another argument in favor of motor trucks! An increase in wages from $5 to $7 a day for 200 members of the Journeymen Horseshoers' Union, with a promise of $1.75 an hour for overtime, has been granted by the Master Horseshoers' Association. As a result of this, the cost of horseshoes has been advanced from $3 to $4 a set for ordinary shoes, and from $1.25 to $1.50 each for patented and bar shoes, and the cost of removing shoes will advance from 50 to 60 cents each.

According to the figures of City Comptroller Nolte, this will mean that the municipal government must pay approximately $8,000 more a year for horseshoes. The street, sewer, health and water departments have 464 horses, and the fire department, 175 horses. The city's bill for horseshoes in 1919 was $25,070.26, which was $1,268.53 more than in 1918.

BRITISH METAL PRICES DROP

LONDON, May 21 (*Special Correspondence*)—Great Britain is in sympathy with America as regards a notable—because so sudden—shrinkage of the price of certain metals. So far the metals affected are copper, tin, spelter and lead, none of them of prime importance to the automobile industry. There is, however, just the possibility of steel, or at least certain classes of it, following suit, and an even stronger possibility that aluminum will be reacted upon by the reduction in copper and spelter.

GRAY PICKS MILWAUKEE SITE

MILWAUKEE, June 7—The Gray Motor Corp., Detroit, has selected Milwaukee as the seat of one of a series of assembling plants to be established throughout the United States. Negotiations are under way for about 40,000 sq. ft. in existing buildings, to be equipped for turning out 10 to 12 cars a day at the start. Local capital is becoming interested to the extent of about $300,000.

German Made Trucks Increase in Price

Exchange Rate Fluctuations Held Responsible for Revisions—Sales Rules Laid Down

WASHINGTON, May 28—Fluctuations in foreign exchange are largely responsible for the price revision upward of German motor trucks. The Association of German Automobile Truck Manufacturers recently fixed a new scale of prices, which reckoned in foreign currency are entirely out of relation with production costs.

This organization has promulgated the following regulations to govern sales:

The domestic price in marks is good for the whole German Empire, the present German Austria, Czecho-Slovakia, Jugo-Slavia (including Serbia), Hungary, Bulgaria and Turkey.

An extra charge of 33 1-3 per cent of the domestic price is made for the former Russian Empire, the present Poland (including Galicia, Danzig, Posen, etc.).

An extra charge of 66 2-3 per cent of the domestic price is made for Finland and the present Roumania, including Siebenburgen. For all other foreign countries prices are fixed exclusively in foreign currency.

Any legally established export duties to be borne by the purchaser. In case of the institution of an official compulsory rate of exchange the seller is relieved of the contract.

Prices for domestic sales, and for German Austria, Jugo-Slavia, Hungary (excluding Siebenburgen), Serbia, Bulgaria, the Saar region and Danzig, provided a guarantee is given that the trucks will remain in the respective countries:

	Marks
Chassis, 2-ton	112,000
Chassis, 3-ton	121,000
Chassis, 5-ton	131,000

Complete, except for tires; according to style of body:

	Marks
2-ton	from 119,000 to 124,600
3-ton	from 129,000 to 142,700
5-ton	from 139,000 to 152,600

For Poland, Russia, Czecho-Slovakia and Danzig, without guarantee of remaining:

	Marks
Chassis, 2-ton	136,000
Chassis, 3-ton	145,500
Chassis, 5-ton	155,000

Complete, except for tires; according to style of body:

	Marks
2-ton	from 144,100 to 150,200
3-ton	from 153,900 to 169,250
5-ton	from 163,000 to 179,000

For all other foreign countries of Europe and overseas and for the Saar region, without guarantee of remaining:

	Marks
Chassis, 2-ton	160,000
Chassis, 3-ton	169,500
Chassis, 5-ton	179,000

Complete, except for tires; according to style of body:

	Marks
2-ton	from 169,000 to 175,750
3-ton	from 178,800 to 195,700
5-ton	from 188,800 to 205,400

Co-operative Housing Proposed by Expert

Employers Get Plan for Solution of Difficulties—Situation Demands Action

ST. LOUIS, June 7—A scheme of co-operative housing is suggested as a possible solution of America's housing problem by Leslie H. Allen of Fred T. Ley & Co., Inc., Springfield, Mass., in a paper on "Industrial Housing—A Financial Problem," read at the spring meeting of the American Society of Mechanical Engineers held this week at the Hotel Statler.

Under the system of copartnership or co-operative housing which is being tried in several New York apartment houses and by the English Garden City Companies, Allen said, a company is organized to purchase and develop real estate and the stockholders are admitted as tenants to the property. He continued:

The usual scheme is for each man to purchase stock to the value of half the cost of his house and land, the other half being carried by a mortgage on the whole property. His rent is sufficient to amortize the mortgage in 7 to 12 years, and then the housing company can either reduce rents or pay large dividends.

To apply this system to the housing of the working classes would necessitate the placing of blocks of stock in the hands of employers and charging enough rent to pay for the purchase of the stock by tenants from the block holders in monthly or annual installments instead of selling the whole of the stock to the tenants at the start. The rent paid would be large enough to take care of installment payments on the stock, as well as the amortization of the bonds. A rental of 12 per cent of the cost of the development would amortize the bonds and place the whole of the stock in the hands of the tenant in about 27 years.

If the employer wishes to retain control he could do so by selling preferred stock to the tenants for their cash payments and dividing with them common stock (of no par value) created for the purpose of controlling the voting power only.

Allen's views concerning the housing problem may be summarized as follows:

Describes Eight Problems

"Our housing shortage is due partly to low rents.

"Partly to an unjustifiable fear in America's ability to make good.

"Houses should be sold to workmen wherever possible.

"Money for financing should be raised locally.

"Payments should be spread out on a long term.

"At least 40 per cent of the cost should be furnished by second mortgages or their equivalent.

"The burden of taxation on houses should be reduced.

"All mortgages should contain an amortization feature.

"Our housing problems can be solved only by action. We should strive to put the business of housing on a self-supporting basis, for we have not solved the problem if our housing does not pay its way. We may make some mistakes in the steps we take, but if all those who are trying to remedy the present state of affairs will work together, we can overcome all difficulties and solve our problems, to the lasting benefit of our country and our homes."

Farmers Ask $250,000 for Fuel Experiments

WASHINGTON, June 5—Shortage of gasoline for motor vehicles in the Northwest has brought a protest from the farmers and other users. Congress has been asked to appropriate $250,000 for experimentation in substitutes for gasoline. Congressman Knutson of Minnesota has introduced a joint resolution for this purpose in an effort to expedite the passage of this measure.

Under the terms of the joint resolution, the money would be turned over to the Bureau of Chemistry, Department of Agriculture, for research. This organization has been conducting experiments at the Arlington farm laboratories near here. They have met with some success in producing a gas from straw and other farm products of similar nature.

The joint resolution has been referred to the House Committee on Agriculture. No action on the measure is anticipated until after recess of the convention. The preamble to the resolution says "the paramount importance of a sufficient supply of gasoline, or a suitable substitute therefor to supply the needs of the American people is a self-evident proposition." The preamble adds "the reserve and visible supply of gasoline is being rapidly exhausted by the growing demand, and the price of which is being advanced to unprecedented levels."

CANADA SHIPS SPARK PLUGS

TORONTO, June 5—It is significant of the steps being taken to readjust the badly distorted balance of trade between Canada and England, that Canadian spark plugs are now being shipped to England for the first time. A large Canadian company has just made its initial shipment of spark plugs, thereby establishing a precedent in the fact that a Canadian manufacturer is shipping into a field which formerly supplied this country, rather than looking to Canada for its supplies.

DEMONSTRATE WITH MOVIES

INDIANAPOLIS, June 5—The Midwest Engine Co., manufacturers of the Midwest utilitor, has supplied all its distributers, dealers and district men with a motion picture machine to show prospective purchasers how the machine operates under all conditions and for all the uses to which it is adapted. The pictures are believed to have greater sales force than the usual more or less perfunctory demonstration.

Allen Creditors Meet Receivers

Factory May Continue Production Pending Financial Rehabilitation—Car Demand Continues

COLUMBUS, June 5—More than $1,000,000 of the $1,400,000 merchandise claims against the Allen Motor Co., which recently went into the hands of receivers, was represented at a creditors' meeting this week. After receiving preliminary reports it was decided to name a special creditors' committee to confer with Receivers George A. Archer and W. C. Willard. This committee consists of R. C. Wolcott of the Miller Rubber Co.; W. J. Cowin of the Columbia Axle Co., and F. D. Whitlinger of the Westinghouse Co. This committee is to report to the creditors within 30 days.

The plans of the receivers as announced by Attorney George M. Hudges is to take a complete inventory of the assets and to keep the plant in operation until such inventory is completed. He said that a refinancing plan is still being considered and that the receivers will announce their policy when the inventory is finished.

C. R. Miller, the general manager, who has been retained by the receivers, said it was the intention to turn out 500 cars monthly in order to use up the stock on hand. He reported a good demand for the car and urged that the plant be sold as a going concern rather than to try liquidation. He claimed that the overhead was being reduced rapidly.

Forest City Company in Receiver's Hands

CLEVELAND, June 7—Upon the petition of creditors, F. W. Treadway has been appointed receiver of the Forest City Machine & Forge Co. The assets are approximately $600,000 and the liabilities about $325,000. Open accounts total $87,171. There are no preferred creditors. According to W. H. Wherry, secretary of the company, it is hoped all debts will be paid in full and the receivership lifted.

The corporation attributes its financial difficulties to the failure of an attempt made last November to float a new stock issue which was designed to carry it over from a war to a peace footing. The stock did not find a ready sale and the trouble because of inadequate working capital was emphasized when banks began to scrutinize credit more sharply a few weeks ago. An attempt to bring new interests into the business was forestalled by the receivership.

MILLER SEEKS COAST SITE

AKRON, June 8—Miller Tire & Rubber Co. is having plans prepared for a factory on the Pacific Coast. F. C. Milhoff, general sales manager, is looking over the ground with the idea of locating a site suitable for a factory with a capacity of 2500 tires daily.

Truck Manufacturers Rush Construction

Demand for Transportation Speeds New Plants in Detroit District

DETROIT, June 7—Some idea of the aid given by the motor truck during the railroad strike and the attendant congestion is furnished by the Duplex Truck Co., which reports a regular line in operation between Lansing, Mich., and Hartford, Conn. The truck trains have been reaching the factory loaded with rims and bearings, and have carried away parts for cars and trucks knockdown, furnishing invaluable assistance on both in- and outbound shipments.

The fact that the truck alone saved the situation is patent to every one who has watched the progress of events in the automotive industry within the last two months. That motorized transportation has become a public necessity has been exemplified clearly.

All factories are using trucks now and have been for some several weeks. Shipments of parts and materials from all sections of the country are being brought into Detroit, and the trucks are returning loaded as heavily as the traffic laws will permit, thus insuring the operation at a minimum cost to the factories.

The attitude of General Motors Corp. is significant. While there has been no slowing up in production of passenger cars it becomes increasingly evident that the real drive of General Motors is to be in the truck end. The additions being constructed at the plant in Pontiac are being rushed to completion with all the speed possible, while other building programs of General Motors are being held back.

Pontiac Plant World's Biggest

The steel has begun to arrive for the immense new machine shop, and within the week will be hoisted in place. While the machine shop is but one of the units being added it alone will add 147,000 sq. ft. of floor space to the manufacturing end. Additions include a new test house, a receiving dock and a storage house for heavy materials. The plan of General Motors is to double the present capacity and make its Pontiac truck plant the largest of its kind in the world. All units will be of concrete and steel, and the machine shop will be joined to the present chassis assembly plant by a two-story section, 80 x 100 ft. The difference in elevation between the machine shop and chassis assembly plant will bring the main floor of the former on a level with the second floor of the latter.

Heavy machinery for the production of trucks, so that the output will not be interfered with, will be placed as quickly as the shop floor is finished. As soon as that is completed the building now occupied by the machine shop will be utilized for heavy storage purposes.

Kalamazoo Motors Corp. also is adding several units to the factory, which will more than double the production of Kalamazoo trucks. The side walls of the assembly building and the roof are in place, and it will be available for use within two or three weeks. The Kalamazoo plant is shipping its smaller trucks, those of the 1½-ton model on the 2½-ton jobs to destination points in various sections of the country. A large shipment started Saturday for Indianapolis, and six 2½-ton jobs now are on the way to Atlanta, Ga., carrying a consignment of the smaller models. A fleet of Kalamazoo trucks went through to Baltimore last week, covering the 674 miles in 6½ days, an average of 12½ miles an hour, arriving in Baltimore in good condition.

Production in May Tribute to Industry

(*Continued from page* 1376)

to the factory as guests of the manufacturers. Entertainment was furnished them at the factories before they started away in their new car.

The next 60 days will see new names and new cars on the road from this district. The Wills-Lee job, description of which is promised within 10 days, and deliveries inside of 60 days, the new Lincoln car, the Handley-Knight and the Jacquet will be moving over the roads as will also the Friend, being produced by the Friend Motors Corp. At Pontiac, the Harroun and the new Saxon.

Manufacturers are not overconfident. They realize the future is problematical and in the light of past events they will make no predictions definitely. The consensus, however, is that increase in production will be constant.

Parts manufacturers in the district and those who supply the automobile factories here also report conditions improving materially and production being speeded up. While there is little indication that conditions will permit a reduction in the cost of parts, it is not believed there will be any further increases for a time at least.

What effect this may have on the price of automobiles cannot be stated. The use of trucks and the long hauls that are necessary in aiding the crippled railroad systems naturally are increasing the cost of production materially and it is possible that another increase in automobile prices will be announced generally within the next 30 days, or around July 1, when many of the companies close their fiscal year and fix their new prices.

DROP FORGE PLANTS GROW

LANSING, MICH., June 5—With the completion of the plant of the Federal Drop Forge Co., Lansing becomes the real drop forge center of the country. The output here will be larger comparatively than that of Detroit or Chicago when the new plant starts production. There now are four plants, the Atlas, Lansing Forge, Melling and Federal, employing about 400 men, and with an estimated output of 15,000 tons a month.

Commission Orders Refrigerator Cars

Automobile Box Cars Give Place to Other Equipment in Railroad Program

WASHINGTON, June 8—Appropriation of $75,000,000 to aid in the acquisition of freight cars as ordered by the Interstate Commerce Commission this week will have little or no effect on the manufacture of automobile car equipment. The Commission has directed that the major portion of this fund now made available under the Transportation Act must be used primarily to aid in the purchase of 20,000 refrigerator cars.

With this preferential treatment for refrigerator cars not much hope is held out for the acquisition of other freight cars. It is generally understood that open top coal cars will be the next on the list of preferences. The commission has made a concession which may allow construction of box cars and special types of equipment used in the automobile industries. It is provided "where carriers offer free of prior liens running gear, parts of cars or types of cars which by construction, reconstruction, or reinforcement may be converted into modern and efficient equipment at an earlier date than is possible by new construction, preference will be given to carriers making such an offer over others tendering the same or substantially the same proportional contributions to meet the loans."

Authorization is given for the expenditure of $50,000,000 for the purchase of locomotives. The commission refused to undertake deformation of an equipment corporation. It intends to give preferred consideration to applications for loans for the purposes for which a corporation may be organized.

WETMORE REAMER TO EXPAND

MILWAUKEE, June 7—The Wetmore Reamer Co. expects to increase its output of expanding and cylinder reamers for the gas engine and automotive industries about 300 per cent after July 15, when it moves into its own plant on Twenty-seventh Street, near St. Paul Avenue. The main shop is 70 x 175 ft., on a 1¼-acre tract. P. B. Rogers is president and general manager; C. P. Wetmore, vice-president and chief engineer, and P. H. Dorr, secretary and sales manager. L. Ray Smith, president of the A. O. Smith Corp., is a director.

HORSE CARS HELP SHIPPER

SPRINGFIELD, ILL., June 5—Robert E. Hatcher, Jr., Studebaker distributer for this territory, has solved the transportation problem by the use of the Arms Palace horse cars. He has been able to get cars all spring and winter, using two cars for the purpose at an expense of $85 additional on each car, as compared with a driveaway cost of $100, besides the damage to the car caused by the driveaway.

Cheap Distribution Is Greatest Need

Production Useless Without Economical Means of Reaching Consumer, Says Expert

NEW YORK, June 8—Advent of the tractor and trailer is one indication that the great industrial need always produces the means to fulfill it, said H. Eltinge Breed, former deputy commissioner of the New York State Highway Commission, in an address delivered at the annual meeting of the National Highway Traffic Association in New York City.

"There are registered in the country to-day 35,000 trailers and their percentage of increase in the last two years has been more than 100 per cent. At this rate of increase, we can plan for 500,000 tractors and trailers by the year 1930, he said. The crux of the economic situation in the country to-day is distribution, even more than production. The motor truck and the tractor and trailer have been developed to meet the needs of distribution. The railroads cannot meet it; industries in the Middle West are being forced daily to shut down because of shortage of freight cars for the transportation of material. Farmers are frantic with anxiety about the movement of their crops.

"Much as we need distribution, we need, even more, economy in distribution. It is appalling to realize that we, as consumers, pay from three to ten times the actual cost of the finished article, be it manufactured or natural product. Whatever effects a saving in cost to the consumer is a national boon. Motor power for short-haul freight does effect such a saving.

"Trailers," continued Breed, "make it possible for one tractor and one driver to do the work of three. Together with the motor truck they have met the industrial need. More and more they are becoming to the great cities the strongest assurance of a sufficient food supply.

Poor Highways Severe Check

"But motor transportation suffers a serious check in the uncertainty of highway design. This spring improved roads have gone to pieces as they never have before. Some people say the damage is due to the hard winter." This, Breed said, he does not believe, maintaining that the roads are organically wrong because they are not designed for the loads they must bear. Unless they can sustain the necessary traffic they do not fulfill their function, but hinder instead of help distribution. The savings effected by motor transportation more than offset the cost of roads that are essential to this transportation.

"Adequate design of highways for tractor and trailer," he said, "means immediate provision for three lines of traffic near all large cities and on all trunk routes, with possibility of an added ten feet longitudinally for a fourth line. Turns must be widened because of the greater length of tractor and trailer, and curves must be superelevated so that the vehicles will keep their own line of traffic. Special attention must be given to reducing grades, because of the increasing cost of fuel. The saving in fuel on heavily traveled main roads often compensates in a year or two for additional cost involved in reducing grades.

"Tractors and trailers are less hard in impact upon pavement than heavily loaded trucks, because of the distribution of load over six or eight wheels and their slower movement. They do not require any greater strength of foundation or better type of surfacing than should be provided for regular heavy traffic, but all heavy traffic requires an adequate foundation and a durable type of pavement; a semi-durable pavement for heavy traffic is sheer waste.

Must Educate Public

"I became more and more convinced," concluded Breed, "that the final success of road building depends upon the intelligence of the public, which must not only demand good roads but must know enough about what constitutes good roads to act as a spur to the conscience of engineers and contractors. Good roads require an alert, intelligent public, engineers with vision, competent contractors and the training in highway work that some universities are now giving. The urgency of the need itself will gradually produce highways that will be adequate to the demands of tractor and trailer."

Engineers to Sponsor Ball Bearing Study

NEW YORK, June 7—At the request of the Swiss Standards Association for co-operation in the standardization of ball bearings, the American Engineering Standards Committee requested the American Society of Mechanical Engineers and the Society of Automotive Engineers to act as joint sponsors for the project. These societies have accepted the responsibility and are now organizing a sectional committee for the work. The sectional committee will be thoroughly representative of all the interests involved and is the body which will be responsible for the detailed formulation of the standards.

HIRED CARS CHEAPER FOR TOURS

NEW YORK, June 8—Prospective European tourists here have been assured that it will be cheaper to hire cars in Europe than to take cars from this country. English companies specializing in this service offer cars for any period at from \$30 to \$40 a day, the renter to pay expense of upkeep and the chauffeur, and from \$37.50 to \$50, with upkeep and chauffeur expenses paid. Special tours exceeding three days are arranged, \$22.50 a day for parties of five in England and about 15 per cent higher on the continent. This includes hotel bills, tips, admission fees, etc. One of the companies in this field is Fraser, McLean Auto Tour & Hire Co. in which E. Malone, who assisted in the entertainment of the American publishers in England, is prominent.

Business Vehicles Head Credit List

Analysis Shows Majority of Cars On Time Payments Bought By Merchants

NEW YORK, June 5—Following publication recently by the General Motors Acceptance Corp. of an analysis of the classes of persons buying motor cars and trucks on the deferred payment plan, the Continental Guaranty Corp. has brought out a comparative analysis of the General Motors study and one undertaken by the Continental two years ago. The comparisons show that the type of buyers of automotive vehicles on time has changed little since 1918. In the earlier analysis the average monthly income of purchasers of cars under the deferred payment plan was \$234 and in 1920 it was \$277, while the average monthly payment has increased from \$51 to \$60. The average of the initial payment also has advanced, being 43 per cent of the cost this year, as compared with 40 per cent two years ago.

The Continental analysis was based on an investigation of sales of 9868 passenger cars while that of the General Motors covered 7891 cars and trucks. The substantiality of the purchaser in both cases is shown by the statement that real estate controlled by the purchaser investigated at the time of 1918 analysis averaged \$7,475, while in 1920 the average holding was \$7,805.

Both analyses showed the majority of vehicles purchased were used chiefly for business purposes. In the Continental analysis 70 per cent of the cars were used chiefly for business purposes and in the recent General Motors analysis 61½ per cent.

Lawson Air Service Delayed to August 15

MILWAUKEE, June 7—Transcontinental passenger service by aerial liners capable of carrying 26 persons, will be instituted about Aug. 15 between New York and Chicago by the Lawson Airline Co. of Milwaukee, which is the pioneer builder of commercial aircraft in the United States. The service originally was intended to start July 1 or 15, but it will not be possible to complete the first ships in time.

A larger plant has been established at South Milwaukee, and materials purchased for an initial lot of ten ships. Several changes in the original design were made, including larger motive power, but no radical departure has been necessary. Some delay was encountered in getting prompt delivery of materials, but Lawson said to-day it is now possible to guarantee delivery of the first ship by July 15 and the remainder at intervals of about two weeks, enabling the operating company to establish regular service by the middle of August. Later it is intended to establish a similar service between Chicago and the Pacific.

Gasoline Shortage Comes Under Probe

California Situation May Lead to Wide Investigation— Would Stop Exports

WASHINGTON, June 10—Instructions were issued to-day by Attorney General Palmer to the United States District Attorney for Northern California to investigate the charges by the California Automobile Trade Association that certain oil companies have created an artificial shortage of gasoline and have already a rationing system. The attention of the Department of Justice was directed to the charges after Senator James D. Phelan had consulted Secretary Daniels, of the Navy; Secretary Payne of the Interior, and officials of the Bureau of Mines.

In its communication to Senator Phelan, the California association alleged that the artificial shortage of gasoline is costing California merchants millions. The effect on the automobile trade is becoming more evident as prices of gasoline are advanced. Government agencies have been asked to investigate the necessity of an embargo on gasoline and oil exports on the Pacific Coast.

The attorney general was advised that the complaint was entered on behalf of the three thousand members of the California Automobile Trade Association. These business men point to the fact that exports continue to increase as the gasoline shortage becomes more alarming. It is openly charged that the oil companies of California brought about the shortage in anticipation of advancing prices to the consumer.

Numerous instances of the hardship imposed on the automobile and other trades by this procedure have been cited. One large firm, the attorney general was told, recently shut down for a week because they were refused oil supplies. With the Pacific Coast taking the initiative in the matter it is understood that a nation-wide investigation far more intensive than the one conducted by the Federal Trade Commission will be instituted.

California Asks Gasoline Probe

FRESNO, June 5—Expressing its belief that the present gasoline shortage in California is artificial and the possible forerunner of a price increase, the California Automobile Trade Association at its annual meeting to-day, called upon Attorney-General Palmer to investigate the situation in an effort to afford relief to the automotive business of the Pacific Coast. The two California senators and eleven representatives were also urged by telegraph to get behind the movement.

One dealer stated that a certain paint shop in Los Angeles has painted 1,000 filling station price signs for the Standard Oil Company quoting gasoline at 30 cents. It is now 23½ cents. This assertion was followed by the suggestion that the propaganda concerning the alleged shortage may have been preparatory to a price increase.

Another dealer, speaking on the subject of a shortage of tank cars, said nineteen cars were spotted beside a refinery and that only one was filled and that this one stood four days on a siding before it was moved. Another dealer told of a car of gasoline which paid demurrage for four days on a siding while the surrounding country was crying for fuel. From many spots throughout the State come statements and rumors about oil wells which are capped, holding the crude oil in the earth.

G. M. C. Within Reach of 1920 Car Schedule

NEW YORK, June 5—Despite railroad tieups and other difficulties which have been hampering automobile manufacturers, General Motors Corp. has been keeping well up to its 1920 production schedule, which calls for 612,000 vehicles. It produced 188,900 passenger cars, motor trucks and tractors in the nineteen weeks ended May 22, compared with 140,900 in the corresponding period last year. This is an increase of 48,000, or nearly 35 per cent. Output thus far in the current year has been at the rate of nearly 10,000 vehicles a week, compared with 7415 a week in 1919.

Since the latter part of 1919 production of Buick cars has averaged about 500 a day. By July the completion of additions to the plants at Flint and St. Louis will permit production of 700 Buicks a day. Chevrolet output of about 800 cars and trucks a day will shortly be increased to 950 a day. With Oakland soon to be on a daily production basis of 250 cars a day, Oldsmobile turning out 500 a day and Scripps-Booth 100 a day, together with the increasing output of trucks and tractors, especially in the Samson division, production of 12,000 vehicles a week by General Motors Corp. is not far distant.

FORM WORLD AIR BOARD

BOSTON, June 5—Major Charles S. Glidden of this city, but now in London, sent a cablegram last night announcing formation of a world's board of aeronautical commissioners, consisting of representative men of 60 countries and colonies, to advance aviation and encourage aerial navigation in all parts of the world.

The officers of the board are: President, Colonel Demont Thompson, New York; executive secretary, Major Glidden; treasurer, Dr. A. L. Hipwell, Paris.

Major Glidden said plans had been completed for a tour around the world in an airplane as a test flight for the projected aerial derby. The plans call for three passengers to be selected by the Aero Club of America, in addition to the crew. Major Glidden estimates the distance to be traveled as 22,000 miles and the flying time as 250 hours.

Petroleum Officers Study Oil Abroad

Effect Here of European Demand Subject of Investigation— Develop Alcogas

NEW YORK, June 9—Acute shortage of gasoline on the Pacific Coast and fear that it may be extended to other parts of the country when the touring season is at its height this summer has served more effectively than anything which has happened thus far to center attention upon the fuel situation.

All the interests in the automotive industry are co-operating so far as possible in seeking a solution of the problem. In this work they are having the support of the oil industry. Thomas A. O'Donnell, president of the Petroleum Institute of America, and Van Manning, former director of the Bureau of Mines but now associated with the Institute, sailed Saturday for Paris. They will attend the organization meeting of the International Chamber of Commerce, after which they will study the oil situation in Europe so far as its bearing upon the United States is concerned.

R. L. Welch, secretary and general counsel of the institute, will deliver an address on the petroleum problem at the summer meeting of the Society of Automotive Engineers at Ottawa Beach late this month. Apparently every effort is being made by oil producers to obtain all the gasoline possible from the sources of supply now available while they are engaged in prospecting for additional wells.

The United States Industrial Alcohol Co. is completing exhaustive experiments with alcogas as a fuel for internal combustion engines. These experiments have been aimed chiefly at determining whether this new fuel is available for extended use in the engines now installed in motor vehicles or whether changes will have to be made in them. It has been understood that about 80 per cent of alcogas was composed of petroleum and its by-products. There are reports that when gasoline reaches 40 cents a gallon alcogas will be put on the market as a substitute.

If gasoline rationing becomes necessary generally, trucks undoubtedly will be given the preference as they have been on the coast. If the fuel supply is augmented by some satisfactory substitute it now seems probable that use of the substitute will be forced as far as possible in passenger cars for there is no intention of permitting the usefulness of trucks to be circumscribed in any way with transportation conditions as they are now.

UNIVERSAL STEEL BUYS SITE

HURON, OHIO, June 7—A steel plant to employ 350 men will be erected here by the Universal Steel Co. of Cleveland. The plant will be in operation within six months, according to F. W. Metter of the Universal company. The factory will be built on a 50 acre tract south of the city.

Employee and Joint Council of the Clark Equipment Company

Problems affecting employer-employee relationship which arise at the plant of the Clark Equipment Company, Buchanan, Mich., are settled in council. Six of the group here pictured are salaried men, the rest wage earners. Eugene B. Clark, president of the company, is ninth from the right, standing

Uniontown Races Draw Select Field

Leading Drivers Will Again Demonstrate Practicability of Small Engines

NEW YORK, June 8—Another test of the 183 cu. in. engines developed by American designers for automobile racing cars will be given on the track at Uniontown, Pa., June 19. The three Monroes and the three Frontenacs designed by Louis Chevrolet, which were driven to fame on the Indianapolis Speedway May 31, all have been entered as well as the three Duesenbergs, which were within the money.

Gaston Chevrolet will drive the Monroe in which he won the Indianapolis classic. Another Monroe driver will be Roscoe Sarles, who was forced out early in the sweepstakes with a broken steering gear but later took the wheel for Bennie Hill in a Frontenac and again was forced out with a broken steering knuckle when he hit the wall on a turn. The other members of the Chevrolet team have not been announced.

The Duesenbergs will be piloted by Jimmy Murphy, Tommy Milton, holder of the world's speed records, and Eddie O'Donnell. Milton finished third and Murphy fourth at Indianapolis. O'Donnell went out near the end of the race.

Only fifteen entries will be permitted at Uniontown and a new track record is predicted for the race. The cars at Indianapolis averaged more than 90 miles an hour until the last fifty miles and they did not by any means reach the limit of their speed. Keen interest is being displayed in the contest not only by race followers but by engineers.

There is manifest a revulsion of feeling against the big engines which consume large quantities of gasoline, especially in view of the present shortage. The smaller engines have not had a fair test in this country and they are expected to improve with experience. Duesenberg has gone farthest in developing them and three of his four entries finished within the money at Indianapolis. While only two of the Chevrolet entries were prize winners it was significant that he designed only the engines and they stood up perfectly under the long grind. The only defects were in other parts of the car.

Louisiana Bill Would Cut Truck Capacities

BATON ROUGE, LA., June 8 — No motor truck with a carrying capacity of more than four tons and no truck or trailer of any description with steel tires will be permitted to operate in Louisiana if a bill introduced in the Louisiana State Legislature Monday night by Representative Nunez of St. Bernard Parish is enacted into law, as it seems likely to be from the influences behind it.

The measure was rushed to second reading Tuesday and was referred to a committee. Automobile men of the State have wired protests to the legislative committees and are preparing to come here in force to combat the proposed law, which these dealers claim is an effort on the part of the railroads to put the ship-by-truck movement in this part of the South out of business.

STEPHENS GETS EXPORT TRADE

FREEPORT, ILL., June 7—Cars manufactured by the Stephens Motor Car Co. went to fourteen foreign countries during the month of May, the largest export business yet enjoyed by the management. The countries served included New Zealand, Syria, Switzerland, Sweden, South Africa, Natal, Norway, Japan, Hawaiian Islands, Guatemala, France, England, Cuba, Brazil and Argentina. R. M. Tarr of East London, Cape Colony, South Africa, representing Malcomess & Co., visited the Stephens factory and closed a contract as distributer for South Africa. The entire output for the year is now sold up.

Mercer to Double Present Production

Plant Additions Insure Completion of Year's Schedule—Earnings High to May

NEW YORK, June 8—Forty thousand shares of the common stock of the Locomobile Co. and all the $5,000,000 stock of the Simplex Automobile Co., owned by the Mercer Motors Co., are carried on the April 30 balance sheet of the latter concern at $1 each. The current market value of the Locomobile stock alone is $400,000. In addition, the balance sheet shows 50 per cent of the stock of Hare's Motors, valued at $750, and 100 per cent of the stock of the Mercer Distributing Co., valued at $500, making a total of only $1,250 for the entire subsidiary holdings.

A statement coverings operations for the first four months of the present Mercer company's existence, ended April 30, showed net earnings before Federal taxes of $203,410, being at the rate of $610,230 for the full year, or better than at the rate of $6.10 a share on the outstanding stock.

Emlen Hare, president, in regard to the company's ownership of Locomobile stock, says in the report to stockholders:

"It will be noted during this period your company has acquired 40,000 shares of the common stock of the Locomobile Co., which stock is estimated to have an earning capacity during 1920 of $7 a share.

"The erection of our additional plant buildings, the installation of new machinery and provision for the necessary small tools, etc., are sufficiently near completion for me to feel that your company will be able to successfully complete the manufacturing program for this and next year, which will mean that during the remainder of 1920 our production will be at double the present rate, and triple the rate during 1921."

Tribute to Haynes Paid by Apperson

Given Credit for Conception of First Automobile—Appersons Built the Car

KOKOMO, IND., June 9—What seemed likely to develop into a perpetual debate over the respective parts played by Elwood Haynes and the Apperson brothers in the development of the first automobile, apparently has been decided. Full credit for the conception of the first gasoline car is given to Haynes by Edgar Apperson. It was built by Appersons.

"The public has somehow gotten the erroneous impression that there has been a controversy in this 'first car' matter," said Apperson. "While I shall point out the value to the automobile world of the work of my late brother, Elmer Apperson, in the engineering and building of the first car, nevertheless the full credit for the germ idea and the initial conception of the automobile as a whole belongs to Elwood Haynes.

"The furthest thing from my mind would be in any way directly or indirectly, to deprive or take from him the conception of America's first automobile. How Mr. Haynes got his first idea and how we served him by working out the mechanics and engineering of the car, is, I believe, an interesting story. Back in the early nineties, Mr. Haynes was associated with the Natural Gas Company of Portland, Ind. His duties required him to do a great deal of cross-country driving with a horse and buggy.

"One day the idea flashed into his mind that a vehicle driven by an internal combustion engine would possess tremendous commercial value. For a year or more, Mr. Haynes turned this idea over and over in his brain—the idea from which sprung America's automobile industry.

Make Idea Practical

"Finally Mr. Haynes was ready to have his idea developed in a practical manner. At that point he needed the services of men who were not only practical machanics but who could work out original engineering problems. For that reason he came to my brother and myself who, besides being mechanics, offered the necessary construction facilities in the shape of the Riverside Machine Works which Elmer Apperson owned and operated personally at that time.

"It must be remembered that every functioning unit in that pioneer car represented a brand new problem. The actual mechanical work was done entirely by my brother and myself and our employees. Mr. Haynes purchased the speed transmission which included four individual clutches of the ordinary lathe type and we Appersons adopted these clutches so they would do the work in the car. Mr. Haynes likewise bought the motor, which was of the small launch type. This he secured in Grand Rapids.

"The car was provided with compensating gears which enabled the outside rear wheel to revolve faster than the inside wheel when the car turned a corner. That little car was given its trial trip on July 4, 1894, and actually ran at a speed of six or seven miles an hour. Many of the fundamental ideas and mechanical actions of that pioneer car are in use in all cars of to-day, only they have been more highly developed.

"So that's the way American automobile manufacture started. And it should be remembered that nowhere else in the world has the automobile industry bulked so big and important as in the United States. Nor has any industry ever developed so rapidly or had such a profound effect on the country as a whole.

"When you think of the present day significance of the automobile it is interesting to recall America's first gasoline car as conceived by Mr. Haynes and built by the Apperson brothers."

Now that this phase of the subject has been definitely settled, it remains to be determined whether the automobile idea originated with Haynes or with Charles E. Duryea, who contends that he was the pioneer and has gathered much data to support his claim. Haynes' position in this respect is that no matter what experimental work Duryea may have done, the car built by the Appersons was the first one exhibited in public and therefore the first to be used.

Glidden Tour Renewal Postponed for Year

NEW YORK, June 8—The renewal of the Glidden tour which the Lincoln Highway Association proposed to stage this summer has been abandoned. Only five entries had been received by the American Automobile Association and the fees have been returned. It was found that manufacturers had neither the time nor the facilities to develop cars for the tour and for that reason it was thought best to defer the contest for another year. There is every probability that it will be held in 1921.

While manufacturers expressed a desire to foster the tour so far as they could they have all along felt this was an inopportune time to renew the contest because of business conditions. They have been unable to make enough cars to fill the orders on their books and were not disposed under the circumstances to tie up plant facilities in building special cars for the tour.

VELIE OUTPUT $30,000,000

MOLINE, ILL., June 7—According to the Velie Motors Corp. the value of this year's output will exceed $30,000,000 if the present rate of production is maintained during the remainder of the year. From 60 to 75 cars are being turned out each day, the majority being driven overland to their destination.

O. E. MANSUR DIES

MOLINE, ILL., June 7—O. E. Mansur, director of the Velie Motors Corp., died suddenly in this city from pneumonia, aged 53. He had been with the Velie corporation for eighteen years.

Gasoline Demand Depletes Reserve

Bureau of Mines Statistics Show Shorter Supply Despite Increased Output

WASHINGTON, June 8—Refinery statistics on petroleum products for March and analysis of the production and consumption of refined oils for the first quarter of this year, shows conclusively that the demand for gasoline is making heavy inroads into the nation's reserve stocks. The Bureau of Mines survey makes it clear that remarkable success has marked a nation-wide effort to increase the supply of gasoline. The returns show, however, that increased production has not been sufficient to bridge the gap created by the constantly increasing use of this valuable by-product. Stocks of gasoline were 14½ per cent greater in March than for the corresponding period of last year. Domestic consumption increased 33 per cent during the same time.

The demand for oil and its products became so great that refiners were obliged to draw upon stocks of crude oil to the extent of 3,373,000 bbl. during this period. The increase in the output of gasoline was made possible by improved refining processes. The Bureau of Mines records show that 265 refineries with a daily capacity of 1,560,345 bbl. of crude petroleum were operating April 1. This compared with March of last year, a decrease of twelve plants and an increase of 317,680 bbl. were noted.

It is of particular interest to the automotive industry that the production of lubricating oils during March is reported to be the largest in three years. The daily average amounted to 2,639,322 gal. In the first three months of the year a decrease of 7,000,000 gal. in stocks compared with the first quarter of last year, domestic consumption of lubricating oils increased 39 per cent or 38,000,000 gal. Shipments and exports exceeded last year's figures by 30,000,000 gal. or 40 per cent.

The comparative analysis of consumption shows that the domestic use for the first quarter of this year amounted to 734,044,807 gal. In the corresponding period of 1919 it was 550,112,468 gal. The stocks of gasoline on hand March 31, 1920, were reported at 626,393,046 gal. Last year the first quarter's stocks were 546,062,429 gal.

OHIO AIR LINE TO START

LOUISVILLE, June 8—Plans for the operation of a passenger and freight service by air between Louisville and Cincinnati were definitely formulated by promotors this week by filing articles of incorporation for the Ohio Valley Aero-Transport Company with capital stock of $40,000, divided into $50 shares. It is planned to have several hydroplanes ready for operation not later than July 15, according to the promoters. Seventy-two Louisville business men and firms have subscribed for stock.

INDUSTRIAL NOTES

The New Era Mfg. Co., Bristol, Conn., has virtually completed its new plant at Meriden, which will provide eight additional acres of factory space. Charles M. Gearing has been appointed division manager. The company is a part of the General Motors organization.

Paris Mfg. & Engineering Co., Paris, Ill., has been organized with a capital of $50,000 to make automotive parts. Officers are R. S. Lloyd, president; Horace Link, vice-president; Henry Crede, secretary and treasurer, and R. W. Hayes, chief engineer and manager.

Harley-Davidson Motor Co., Milwaukee, which is erecting a six-story factory addition providing about 200,000 sq. ft. of floor space, also is starting work on a new stock house, 50 x 225 ft., at the main works, Thirty-seventh and Chestnut streets.

Lavine Gear Co., Milwaukee, is making a $100,000 works addition. The company manufactures steering gears and other parts for motor cars, trucks, tractors, etc. The addition is 150 ft. square and will increase the capacity about 75 per cent.

Miami Tractor & Mfg. Co., Celina, Ohio, has been incorporated with a capital of $200,000 to manufacture tractors for farm purposes. The company will take over a tractor which has been in process of development for some time.

Reed-Prentice Co., Becker Milling Machine Co., and Whitcomb-Blaisdell Machine Tool Co., have opened a sales office in the International Machinery Exhibit, Grand Central Palace, New York, for business in that territory.

The Beloit (Wis.) Iron Works has started work on a brick and gray iron foundry addition, 117 x 160 ft., costing about $100,000 fully equipped. It will be ready about Aug. 1.

Continental Motors Corp., has just closed a deal for 170 acres of land on the Muskegon River near here, on which it will erect a foundry. Power is to be developed from the river.

Greenfield Tap & Die Corp., Greenfield, Mass., has bought up the common stock of the Lincoln Twist Drill Co., Taunton, Mass. This gives the Greenfield corporation a complete line of small tools.

Westinghouse Electric & Mfg. Co., has transferred its commercial and order parts of the meter, fan motor and rectifier stations, supply department, from the East Pittsburgh works to the Newark works.

Oakland Motor Car Co., Pontiac, has moved its offices into the new administration building. The former office space will be used for manufacturing.

Winther Motor Truck Co., Kenosha, Wis., has broken ground for a large addition to its present plant which will be ready for occupancy by about Aug. 30.

Barco Battery Co., has been formed in Detroit, to make an automobile storage battery. The new company is headed by P. B. Williams.

BOSCH RE-ENTERS PARTS FIELD

NEW YORK, June 7—The American Bosch Magneto Corp. has just announced a line of automobile generators and starters. The predecessor of the present Bosch concern, the Bosch Magneto Co., some time before the war developed an electric generating and starting system, and later on bought out the Rushmore Dynamo Works of Plainfield, N. J. Both the original Bosch and the Rushmore systems were offered to automobile manufacturers for a time but after the reorganization, probably on account of the heavy demand for magnetos for motor trucks, the line of generators and starters was dropped.

The generators of the new line are to be furnished in two sizes, of 4 in. and 5 in. diameter respectively, and will be wound for either 6 or 12 volts. The third brush system of regulation will be used. A technical description and illustrations of the generators and starters will appear in an early issue.

Hudson Tire Buys Plant Site in Yonkers

YONKERS, N. Y., June 7—The Hudson Tire & Rubber Corp., recently incorporated with a capital of $1,000,000, has purchased a tract of land in this city upon which to erect a factory for the manufacture of all kinds of tires and tubes. W. M. Doucette is president and general manager of the company and H. B. Seymour is vice-president and treasurer.

Doucette started in the bicycle and carriage tire business 25 years ago and has held responsible positions with several tire companies. He resigned recently as Eastern district manager for the Mason Tire & Rubber Co. Seymour has had considerable experience in the tire business and for the past three years has been connected with a rubber and tire trade journal.

Stutz Stock Dividend to Be Paid in Lump

NEW YORK, June 9—Directors of the Stutz Motor Car Co. decided at a meeting yesterday that the stock dividend of 80,000 shares which was to have been distributed in instalments up to April 8, 1921, would be paid in a lump amount on June 29 to holders of record of June 18. Since the corner which resulted in the stock being stricken from the list of the Stock Exchange it has been traded in to a limited extent on the Curb but there has been little fluctuation in price. The announcement of the decision in regard to the stock dividend was in connection with a declaration of a cash dividend of $1.25 a share payable July 1 to holders of record June 15.

HUNT TAKES OVER FOUNDRY

BOSCOBEL, WIS., June 7—The Hunt Mfg. Co. has taken over the Ruka Foundry & Machine Co. plant in order to increase its production of a "jiffy" mandrel for automotive repair shops to 75 a day. The tool has been on the market only a few months, but orders have been received from jobbers and dealers in nearly every State. Repeat orders are of such volume that the Hunt company is obliged to take over the entire capacity of the shop, which has been manufacturing the device on contract up to this time.

METAL MARKETS

Iron and Steel—Sour grape talk galore is being fed to Pittsburgh steel market correspondents by sales managers who have received cancellations from automotive interests. Cancellations and suspensions are a time-honored trade custom in the steel trade and those steel interests that are now suffering from the voiding of orders by automotive buyers have in a large measure only themselves to blame, for a few weeks ago the talk centered around the automotive industry absorbing steel that should have gone to the railroads. One market reporter claims to see that the steelmakers will take their cue from the Federal Reserve Bank and will be less inclined to bid in the future on steel tonnages required for passenger cars. Such comments appear all the more absurd when it is remembered that the steel industry to-day is over-equipped as the result of the tremendous expansion brought about through the war's exigencies, and it is largely the automotive industry that furnishes an outlet for productive equipment that would otherwise be useless. In the Youngstown district, sales managers are freely discussing the change that has come over the market since the automotive industry has shown less eagerness for material. There is no longer a line of purchasing agents waiting to outbid one another in the way of premiums for early deliveries. In fact, some sheet orders have been cancelled. Most of the sheet mills are, however, still busy, chiefly converting sheet bars for account of automobile manufacturers. With the production schedules of automotive builders disarranged by the inadequate shipments of steel during May, specifications against orders for bolts and nuts have become much lighter. As much steel was held in May in unfinished form, the ensuing few weeks may witness quickened activity at finishing mills without a corresponding enhancement in the output of raw steel.

Pig Iron—When for one reason or another, output of the blast furnaces declines, consumers are never left to want for the minutest details regarding the causes and extent of the curtailment, but very little pains is ever taken to awaken them to a realization of marked gains in production. In the last few days the tide in the pig iron industry has swung decidedly the other way, recovery in production having been as spectacular as was its shrinkage a few weeks ago. Coke is still subnormally light in supply and abnormally high in price, but even in that commodity improvement in production is sufficient to make continuance of the nominal price level of $15 for any length of time dubious. Odd sales of steel making grades of pig at relatively high levels have had no bearing on the market for foundry iron, the base price of which remains at $45, furnace, with the undertone rather easier.

Aluminum—The market rules quiet with the sole American producer holding to 33c. and "outside" offerings of virgin ingots, 98 to 99 per cent pure, reported at ½ to 1c. below that level.

Tin—As is always the case in a declining tin market, there are no buyers. At times the price for Straits breaks lower than 50c., but that level appears to be the goal of the price pendulum's swings.

Copper—Domestic demand is still lacking and at times sales of electrolytic at as low as 18⅛c. are heard of.

Lead—The market for lead has turned quiet but the tone is steady to firm.

Zinc—Buyers show very little interest and quotations are nominally unchanged.

Automotive Financial Notes

Allis-Chalmers Co.'s action in declaring an initial dividend of 1 per cent on the common stock of the company has in the judgment of Wall Street placed that stock on a 4 per cent basis. There was no official announcement, however, that the dividend would be paid each quarter. The element of surprise in the action of the board was shown by the market action of the stock which moved from 35 to 38¾ and closed 3⅛ points higher at 38⅛.

Prudential Tire & Rubber Co. of Erie, Pa., which has taken over the plant and equipment of the Boone Rubber & Tire Co. at Chippewa Falls, Wis., has filed articles in Wisconsin, giving its capital at $4,000,000 preferred and 60,000 shares of non-par valued common, and a $100,000 interest in this State. George N. Graham, general manager of the newly acquired works, is designated as agent.

American Motorcycle Co. has purchased the Kentucky Laundry Co., and will convert it into a motorcycle factory. The cost was $30,000. E. D. Hatcher, Otto Seelbach, and J. C. Murphy, are the incorporators of the American Motorcycle Co., which has $200,000 capital stock. It expects to begin making motor cycles by Sept. 1.

Longdin-Brugger Co., Fond du Lac, Wis., manufacturing special closed tops for open passenger cars, has moved into its new factory. The capitalization is being increased from $120,000 to $240,000 to finance increased production, equipment, etc. The bodies are marketed under the name of "Close-Tite."

Stewart-Warner Speedometer Co. stockholders have approved the increase in stock from 400,000 shares to 600,000 shares of no par value. Part of the additional stock will be used for exchange of Stewart-Warner stock and the balance will remain in the treasury for such future use as the directors may determine.

Iron Mountain Co. has completed a large and modern factory in Chicago, for the manufacture of worm drive axles for motor trucks. C. E. Jernberg is president of the company and H. L. Knudson, chief engineer. The axle which is being built is the result of careful study and experimentation.

Republic Motor Truck Co., Inc., for the quarter ended Dec. 31, 1919, reports as follows: Net profits after expenses $530,799; other income $100,792; total income $631,591; charges $138,486; net profits $493,105; net profit of Torbensen Axle Co. and Powrick Co. $132,891; total net profit $635,996.

Ajax Auto Lock Co., West Bend, Wis., organized to manufacture locks and safety devices for motor vehicles, has been incorporated with a capital stock of $50,000 by R. C. Labisky, L. Kuehlthau and F. W. Bucklin. It is now starting quantity production.

Grant Motor Car Corp.—Balance sheet as of April 30 shows total assets of $6,081,687. This includes $228,387 in cash, $515,372 in notes and accounts receivable and $1,650,611 in its inventory. The surplus amounts to $266,000. Notes and accounts payable total $523,790.

Maibohm Motors Co. directors have declared the regular quarterly dividend of 2 per cent upon the capital stock of the company, payable July 1, 1920, to stockholders of record June 15.

Fulton Co., Milwaukee, manufacturing automotive equipment, having completed a new works in West Allis, has increased its capital stock from $75,000 to $150,000.

Joseph Obenberger & Sons Co., operating a forge works, Milwaukee, has increased its capital stock from $25,000 to $50,000.

Royal Tire & Rubber Co., Boston, has been incorporated with a capital of $50,000, for the manufacture of tires.

Centrifugal Shock Absorber Co., Lynn, Mass., has been incorporated with a capital stock of $250,000.

Alliance Top Body & Trimming Co., Boston, has been incorporated with a capital stock of $50,000.

Michigan Oldsmobile Co., Cleveland, has decreased its capital stock from $500,000 to $1,000.

International Truck Enjoys Record Quarter

NEW YORK, June 8—International Motor Truck Co.'s earnings in the first four months of this year exceeded $1,300,000 after charges and taxes, or at the annual rate of about $10 a share on the 283,108 shares of common stock. The $8,000,000 assets which International Motor Truck received through its consolidation with Wright-Martin Aircraft Co. are just beginning to show results.

Profits last year were equivalent to about $25 a share on about 70,778 shares of common stock outstanding Dec. 31, 1919. As a result of its consolidation International Truck now has outstanding $10,921,000 cumulative 7 per cent first preferred and $5,331,000 cumulative 7 per cent second preferred. It has no funded debt.

Traffic Declares 50 Per Cent Dividend

ST. LOUIS, June 7—A stock dividend of 50 per cent has been declared by the Traffic Motor Truck Corp. on its $500,000 of issued capital stock. The par value of the stock is $100 a share. The company's capital stock was increased to $1,000,000 on Jan. 1 last, the increase being taken out of these earnings of the company and the added stock being placed in the treasury. The $250,000 dividend was taken out of the treasury stock. It was announced that the company had no stock for sale to the public.

BARTON AXLE TAKES AMERICAN

BARTON, WIS., June 7—The Barton Axle Co. has been organized, with $150,000 capital, to take over the interests of the American Axle Co. of Chicago at Barton, Washington County, Wis., where a new machine shop and assembling plant, costing $110,000, is being made ready for production. Chicago, Milwaukee, West Bend and Barton capital own the new corporation. Officers have been elected as follows: President, E. W. Macavoy, Chicago; vice-president and treasurer, Peter C. Wolf, Barton; secretary, George H. Gabel, Milwaukee; directors, W. C. Dayton, Chicago; M. H. Grossman, Milwaukee; Andrew Hauser, Neenah, Wis., S. J. Driessel and H. W. Suckow, Barton. Dayton is chief engineer, and Charles Flitner, formerly of Milwaukee, works manager.

Bank Credits

Written exclusively for AUTOMOTIVE INDUSTRIES *by the Guaranty Trust Co., second largest bank in America.*

NEW YORK, June 10—The advance of the New York Federal Reserve Bank's discount rate for commercial paper from 6 per cent to 7 per cent, effective on Tuesday of last week, has been followed by like advances by the Federal Reserve banks of Boston, Chicago and Minneapolis. All four banks, together with the Federal Reserve Bank of Richmond, in keeping with other increases, have also raised their discount rates on paper secured by Liberty Bonds and Victory Notes to 6 per cent. The Federal Reserve Bank of Atlanta has been added to the list of banks which have adopted the system of progressive discount rates. The standard quoted rates of this group of Reserve banks are, accordingly, only nominal.

The advance of the New York Federal Reserve Bank's discount rate for commercial paper was reflected in the local open market money rates. The rate for 60 to 90 days' endorsed bills and 6 months' choice names rose to 7¾ per cent, compared with 7½ per cent a week earlier, while the range of advances for less well known names was ½ of 1 per cent. Time money rates remain unchanged, with a less abundant supply of money than in the previous week.

The persistent demand for money, despite the increased rates of discount, is reflected in the borrowings last week of member banks at the New York Federal Reserve Bank. The borrowings on commercial paper increased by more than $39,000,000. This condition was aslo reflected in the clearing house institutions, where the item of bills payable, acceptances, etc., rose $55,985,000.

The Federal Reserve banks as a whole made a further gain of about $8,000,000 in gold reserves last week. Total bills discounted, despite reductions in the volume of bills secured by Government war paper and those purchased in the open market, advanced $36,915,000; net deposits rose $12,735,000; and note circulation, $20,270,000.

The officially reported condition of the cotton crop for May 25, 62.4 per cent of normal, comparing with a 10-year average at corresponding dates of 78.7 per cent and 75.6 per cent at this time last year, is the lowest recorded in the 50 years of Government reporting. While later material improvement in the crop may be expected, its present condition is an additional factor making for a slackening of business activity.

Men of the Industry

H. M. Sloan, formerly assistant to the president of the Chicago, Rock Island & Pacific Railroad, has been appointed treasurer of the Buda Co. He entered the service of the Rock Island Railroad in 1902 and rose to the rank of vice-president. He became assistant to the receiver of that road and later assistant to the president. He remained with the road until he went to Washington in 1918, to serve on the War Industries Board.

Glenn L. Orr, one of the well-known figures in the manufacturing world, has been made secretary, treasurer and general manager of the Lansing Foundry Co., Lansing, Mich. Orr has served successfully with the Detroit Engine Works, Hupp Motor Car Corp., Packard and Briscoe. William Grant of Rock Island, Ill., has been named foundry superintendent. Orr succeeds S. B. Spaulding, who tendered his resignation June 1.

Charles P. Grimes, who was in charge of most of the dynamometer laboratory tests for the government during the war has joined the staff of the Franklin Automobile Co., as research engineer. He is now making an exhaustive study of the Franklin air-cooling system. Grimes designed and superintended the construction of the venturi wind tunnel for calibrating airplane speed indicating instruments.

R. H. Allen, of the Dodge organization, has been made director of purchases, succeeding F. J. Haynes, general manager, whose promotion to head the big organization was announced officially last week. Charles W. Matheson, formerly director of service, has been made acting general sales manager.

W. H. Masten, general sales manager of the Oakland Motor Car Co. at Pontiac, Mich., has been moved up to assistant general manager and C. J. Nephler has been named sales manager. Masten and Nephler were honor guests at a luncheon to W. F. Warner, general manager of the company, last week.

C. E. Thompson, president of the Motor and Accessory Manufacturers' Association, and M. L. Heminway, general manager, sailed Saturday for Paris, to attend the organization meeting of the International Chamber of Commerce. After the conference they will study automotive conditions in Europe.

John I. Hoke, of South Bend, Ind., has been engaged by the Moline Plow & Tractor Co., as special tractor advisor. He has also sold his patent rights in a two wheeled tractor to the Moline company. Hoke has devoted most of his life to experimenting with and inventing various types of tractors.

Ivar Du Rietz of Sven Du Rietz & Co., Inc., Sweden, is returning to Sweden after a month and a half visiting manufacturers represented by his company. The company has opened an office in Chicago in charge of Walter E. Devlin so that it may keep in personal touch with factories here.

A. H. Rowan, of the engineering department of the Southern Motor Manufacturing Association, has opened an office in Detroit at 294 East Jefferson Avenue for the purchase of materials used by his organization. The company builds tractors, trucks, automobiles, trailers and bodies.

S. E. Ackerman, sales manager of the Franklin Automobile Co., sailed yesterday for Liverpool to study conditions in Europe. While in London he will address a meeting of all Franklin representatives in Europe. His itinerary will include Paris, Brussels and other cities.

H. W. Schnetzky has been made president and general manager of the Wisconsin Motor Mfg. Co., Milwaukee. He succeeds Charles H. Johns who has retired from active service. A. F. Milbrath will continue as engineer and C. L. Cole as sales manager.

Rapael Bianchi, Oldsmobile distributer in Spain, and his assistant, Ramon Tabias Patchame, arrived in Lansing this week for a visit to the Olds factory. Bianchi is a nephew of the designer and manufacturer of the Italian car of that name.

Byron H. King has been appointed manager of the Atlanta branch of the Buick Motor Co. He formerly was a member of the factory organization at Flint, Mich. Ben. F. Ullmer succeeds King as assistant branch factory manager.

P. K. Dayton will be in charge of the sales office in the International Machinery Exhibit, Grand Central Palace, of Reed-Prentice Co., Becker Milling Machine Co., and Whitcomb-Blaisdell Machine Tool Co.,

H. L. Beckwith, of the General Motors Truck Co. service department, has resigned, effective July 1. S. V. Norton, formerly with the Goodrich Rubber Co., has been named Beckwith's successor.

R. L. Dean, formerly White representative in Kern County, California, has been appointed manager of the Garford Motor Truck Company's Los Angeles factory branch.

G. E. Swartz, formerly mechanical superintendent of the Torbensen Axle Co., of Cleveland, has resigned to become manufacturing manager of the Timken Detroit Axle Co.

Alfred Reeves, general manager of the National Automobile Chamber of Commerce, has been elected vice-president of the Trade Association Secretaries of New York City.

R. Jackson Jones has been appointed assistant general sales manager of the Traffic Motor Truck Corp., St. Louis, it is announced by H. H. Hawke, general sales manager.

S. V. Norton has been appointed service manager of the General Motors Truck Co., effective July 1, to succeed H. L. Beckwith, resigned.

Edward Hassig has been appointed superintendent of the heat treating department of the Lavine Gear Co. M. C. Griswold has been named production manager.

F. I. Laikens has been made publicity manager for the Allen Motor Co. at Columbus. Laikens is an old newspaper man.

A. T. Jackson, vice-president in charge of sales of the Emerson-Brantingham Co., has left on an extended trip through Europe.

F. K. Blanchard has been appointed assistant engineer of the engine division of the Buda Co., Harvey, Ill.

V. C. Kloepper has resigned as chief engineer of the Astra Motors Corp., St. Louis.

Victor Truck Head Held on Fraud Charge

JACKSON, MICH., June 10—Cyrus C. Van Wagner, president and general manager of the Victor Truck Co., Inc., was brought to Jackson this week to answer a charge of fraud in connection with alleged stock sales in this city. He was held under bond for a hearing.

The warrant charges misrepresentation in the sale of stock in the Central City Paint Co. to employees of the Jackson Cushion Spring Co. Van Wagner declined to make any statement, but friends said the warrant was the result of spite work on the part of persons inimical to the interests of the Victor Truck Co.

The Victor Truck Co. is a Michigan concern, capitalized at $150,000, and it is said in St. Joseph plans were being formulated for the sale of the stock of the truck concern when the request for Van Wagner's arrest arrived. Officials of the Victor Truck Corp. refused to make any statement as to the proposed financing plans of the company and contented themselves with the claim that the action of Jackson authorities was prompted by persons who hoped to hamper the new concern.

Morgan New President of McGraw Company

EAST PALESTINE, OHIO, June 5—John Morgan, formerly vice-president and treasurer of the McGraw Tire & Rubber Co., has been elected president to succeed Edwin C. McGraw, who died suddenly at his winter home in Florida. Morgan will continue to serve as treasurer. He has been with the company almost since its inception more than ten years ago and always has taken an active part in shaping its policies.

Morgan was born in London in 1880 and came to the United States in 1905 to introduce a European tire. He is one of the directors of the Rubber Association of America.

Charles H. Wheeler, factory manager for the company, has been elected a director.

Charles H. John Retires as Wisconsin Motor Head

MILWAUKEE, June 7—Charles H. John, president and treasurer of the Wisconsin Motor Mfg. Co., one of the principal passenger car and motor truck engine builders in the United States, has resigned to take a long rest after twelve years' connection with the institution. Hugo W. Schnetzky, a prominent architect and business man of Milwaukee, has been elected president and treasurer. A. F. Milbrath, secretary and chief engineer, and other officers, continue in their respective capacities. Under the direction of John, the Wisconsin company grew from a small machine shop building multiple cylinder passenger car engines on a small scale in 1908 to a $2,000,000 corporation employing more than 1250 men at this time. An enviable record was made during the war in building engines for the F W D truck, the Milwaukee plant being for months engaged in this war work up to 100 per cent of capacity.

Calendar

SHOWS

Aug. 23-27—San Francisco, National Traffic Officers' Safety First Exposition, Auditorium, C. De Witt De Mar, Manager.

Oct. 6-16—New York. Electrical Show. Grand Central Palace. George F. Parker, Manager.

Dec. 10-18—New York. Motor Boat Show. Grand Central Palace.

Jan. 8-15—New York. National Passenger Car Show, Grand Central Palace. Auspices of N.A.C.C.

Jan. 29-Feb. 4 — Chicago, National Passenger Car Show, Coliseum, Auspices of N.A.C.C.

FOREIGN SHOWS

June 21-27—Sunsvall, Sweden. Automobile Exhibition. Norrland Fair.

June 26-July 25 — Commercial vehicles, tractors, camions and engines. Antwerp.

July 9-20—London, England. International Aircraft Exhibition. Olympia. The Society of British Aircraft Constructors.

Aug. 7-Sept. 15 — Motorcycles, sidecars, etc. Antwerp.

October—London. Commercial Vehicle Show, Olympia.

November—London. Passenger Car Show, Olympia.

CONTESTS

June 14—Omaha, Neb. Truck Reliability Run.

June 17—Portland, Ore. Dirt track.

June 17-18—Chicago, Inter-Club Run, Chicago Automobile Club.

June 19—Uniontown, Pa. Speedway.

June 19—Ogdensburg, N. Y. Dirt track.

July 4—Tacoma, Wash. Speedway.

July 4—Hanford, Cal. Dirt track.

July 4—Spokane, Wash. Dirt track.

July 5—Batavia, N. Y. Dirt track.

July 17 — Warren, Pa. Dirt track.

July 24—Watertown, N. Y. Dirt track.

July 31—Fulton, N. Y. Dirt track.

Aug. 7—Erie, Pa. Dirt track.

Aug. 14—Buffalo, N. Y. Dirt track.

Aug. 20-21—Middletown, N. Y. Dirt track.

Aug. 21—Johnstown City, Pa. Dirt track.

Aug. 21—Elgin, Ill. Road race, Chicago Automobile Club.

Aug. 28—Canandaigua, N. Y. Dirt track.

Aug. 27-8—Flemington, N. J. Dirt track.

August, 1920—Paris, France, Grand Prix Race. Sporting Commission Automobile Club of France.

Sept. 1—Glidden Tour—N. Y. to San Francisco.

Sept. 5 — Targa Florio Race. Sicily.

Sept. 6—Hornell, N. Y. Dirt track.

Sept. 6—Cincinnati, O. Speedway.

Sept. 6—Uniontown, Pa. Speedway.

Sept. 17-18—Syracuse, N. Y. Dirt track.

Sept. 25—Allentown, Pa. Dirt track.

Oct. 1-2—Trenton, N. J. Dirt track.

Oct. 8-9—Danbury, Conn. Dirt track.

CONVENTIONS

June 22-25—Asbury Park, N. J. Annual meeting American Society for Testing Materials.

S. A. E. MEETINGS

June 21-25 — Ottawa Beach, Mich. Summer Conference.

S. A. E. Closes Program for Summer Meeting

NEW YORK, June 9—The hotel reservations made by the Society of Automotive Engineers for its summer meeting at Ottawa Beach, June 21-25, already have been exceeded and in point of attendance the session promises to be the most successful in the history of the organization.

Monday afternoon will be devoted to a meeting of the Standards Committee and the business session will be held in the evening. Transportation will be the topic Tuesday evening and the speaker will be W. D. Schultz. Wednesday evening Colonel Thurman H. Bane, of the army air service, will read a paper on research work at McCook Field, which will be illustrated with stereopticon views and moving pictures. Thursday night there will be a dance and an elaborate sport program has been arranged for each afternoon.

The complete and revised program of the meeting follows:

Tuesday Morning—Fuel Session.

Introductory Remarks by Chairman. C. F. Kettering

Intake Manifold Temperature and Fuel Economy. W. S. James, Bureau of Standards.

Discussion.

Report of Automotive Fuel Committee of S. A. E. H. L. Horning, Chairman.

Discussion.

Carburetion and Distribution of Low Grade Fuel. O. H. Ensign.

Discussion.

Paper on Fuel. R. L. Welch.

Discussion.

Symposium on Engine Design. (a) H. M. Crane, (b) C. A. Norman, (c) W. E. Lay and (d) P. S. Tice.

Closing Remarks by Chairman.

Wednesday Morning—Transportation Session

Introductory Remarks by Chairman. J. G. Vincent

Motor Bus Transportation. G. A. Green.

Discussion.

Motor Transport.

Discussion.

Proposed Program of S. A. E. Committee on Science of Truck Operation. F. W. Davis

Discussion.

Air Navigation. Capt. E. S. Gorrell.

Discussion.

Some Inland Waterway Transportation Problems. V. E. Lacy.

Discussion.

Closing remarks by chairman.

Thursday Morning—Farm Power Session

Introductory Remarks by Chairman. F. A. Johnston

Analysis of Fundamental Factors Affecting Tractor Design. O. B. Zimmerman.

Discussion.

The Operating Speeds of Agricultural Implements. Percival White.

Discussion.

Notes on Power Farming. R. W. Lohman.

Report on Minneapolis Section Tractor Research. A. W. Scarratt.

Closing remarks by chairman.

Friday Morning—Production Session

Introductory Remarks by Chairman. Harold Emmons

Production Control and Systems of Accounting. A. G. Drefs.

Discussion.

The Workman, an Element in Production. A. F. Knobloch

Discussion.

Paper by Magnus W. Alexander—Some Important Phases of the Industrial Situation.

Discussion.

Interdepartmental Production Contests. R. R. Potter

Discussion.

BOSTON-NEW YORK LINE STARTS

BOSTON, June 5—The first commercial airplane to arrive in Boston from New York struck the water and glided up to Commonwealth Pier in a cloud of spray after a 3½ hour trip. The flight followed the steamship lane over Long Island Sound, the Cape Cod Canal and Massachusetts Bay. The distance covered was 230 miles. It cost each passenger who made the trip $150, and there were two passengers. Announcement is made that the service will be conducted through the summer on a three-times-a-week schedule.

Guayule Rubber Plants to Resume Operations

TORREON, MEXICO, June 7—Revival of the guayule rubber industry on a big scale is now in sight. Several rubber manufacturing plants in the Torreon region that have been closed down for five to eight years are to be placed in commission as soon as they can be overhauled and the necessary supply of raw material obtained to assure that they may be kept in constant operation. It is predicted that there will be a number of new manufacturing plants built as soon as it is definitely settled that the country is free of revolutions and banditry.

The available supply of guayule shrubs from which the crude rubber is obtained is greater now than at the time the industry was started, more than sixteen years ago. This is due to the fact that the cut-over lands, especially those upon which the shrub was cut off at the roots, have within the last few years reproduced a crop of the brush that is more bountiful than the original growth. Some of the crude rubber manufacturers have also met with much success in seeding vast tracts of open land with the shrub and the crop produced by this means is now ready for harvest, it is stated.

SMALL PLANES IN DEMAND

OMAHA, NEB., June 6—Airplane sales are getting to be common in this territory, as the Ak-Sar-Ben flying field is the scene of daily flights by owners of several private machines, who have gone so far as to fly to neighboring cities on business trips instead of going by train or automobile. The Western Motor Car Co., Curtiss dealers out here, report quite a few inquiries for the smaller planes and expect to make several deliveries in the near future.